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SOUTHWEST RESEARCH INSTITUTE
Post Office Drawer 28510, 6220 Culebra Road
San Antonio, Texas 78228-0510

ON-ORBIT COMPRESSOR TECHNOLOGY PROGRAM
FINAL REPORT

By
Danny M. Deffenbaugh
Steven J. Svedeman
Edgar C. Schroeder
C. Richard Gerlach

Prepared For
NASA Lyndon B. Johnson Space Center
Propulsion and Power Branch
Houston, Texas 77058

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Approved:

Robert L. Bass, Vice President
Mechanical and Fluids Engineering Division

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

EXECUTIVE SUMMARY

This final report presents a synopsis of the On-Orbit Compressor Technology Program performed by Southwest Research Institute for NASA-JSC under Contract No. NAS9-18051. The objective is the exploration of compressor technology applicable for use by the Space Station Fluid Management System, Space Station Propulsion System, and related on-orbit fluid transfer systems. The approach is to extend the current state-of-the-art in natural gas compressor technology to the unique requirements of high-pressure, low-flow, small, light, and low-power devices for on-orbit applications. This technology is adapted to seven on-orbit conceptual designs and one prototype is developed and tested.

The compressor technology development is based on detailed performance modeling and breadboard testing. The comprehensive digital time domain model used as a design tool includes: cycle thermodynamics, real gas properties, in cylinder heat transfer, valve dynamics, ring leaks, piston friction, attached piping acoustic effects, and piston inertial effects. The design concept is an eccentric crankshaft with anti-friction bearing driven pistons and actuator return springs. Self-lubricated guide and seal rings are used to center the piston and maintain gas pressure. The driving means is a variable speed electric motor. Active cooling is used at the cylinder walls and pulsation bottles.

The prototype compressor developed under this project is intended for compressing oxidizing gas mixtures for storage/disposal. This waste gas comes from experiment modules and is discharged into a vessel initially at 100 psia. The compressor pumps the vessel up to 1000 psia. The prototype design is a 3-cylinder, two stage reciprocating piston type compressor with pressure actuated check valves. The total displacement is 0.625 cubic inches. The unit weighs 30 pounds and fits into an envelope of 0.5 cubic feet. The maximum power requirement is 500 watts and the estimated life is 9500 hours.

The overall conclusion is that compressor technology has been developed for on-orbit applications that balances all of the complex design requirements, and is provided within a time frame consistent with the Space Station Freedom schedule. The test program has documented that the performance of the prototype compressor meets the EIS requirements.

1.0 INTRODUCTION

Space Station Freedom is the first on-orbit application requiring mechanical gas compression devices. All prior on-orbit applications have utilized pressurized tanks to supply high-pressure gas. On-orbit applications constitute a unique set of design requirements that cannot be met by existing gas compressor technology. This unique combination of requirements is for high-pressure, low-flow devices with minimum size, weight, and power along with the need for low maintenance, long life, and component commonality between compressor applications.

The objective of this project, therefore, is the exploration of compressor technology applicable for use by the Space Station Fluid Management System (FMS), Space Station Propulsion System, and related on-orbit fluid transfer systems. The approach is to develop conceptual designs for seven on-orbit applications, develop a detailed design of a prototype for one of these applications, fabricate this prototype and verify it's performance. As a result of this technology development effort subsequent projects will translate this technology into flight hardware in a time frame consistent with the support of Space Station Freedom and future on-orbit missions.

The starting point for this Technology Development Program was Southwest Research Institute's 30-year history in compressor technology for the natural gas industry. The state-of-the-art (SOA) for this technology is based on large, heavy, high-pressure, and high-flow natural gas compressors. This project concentrated on adapting this SOA to the unique requirements for on-orbit applications. As a part of this effort, a prototype unit for one on-orbit application was designed, built, and tested. During the course of this project, however, a number of significant changes occurred in the Space Station program. These changes included a continual evolution of Space Station requirements, a continual evolution of the Fluid Management System operating conditions, and a new requirement for endurance testing of the prototype at NASA-JSC. These affected the End Item Specifications for a number of the applications, as well as the prototype.

Specific modifications required an investigation of: (1) trace contaminants, (2) life extension (increase in life goal by a factor of 10 beyond current SOA), (3) compressor vibrations, and (4) design changes due to new operating conditions.

Since this is a Technology Development Program, the significance of the results from this project can be applied to the ever changing technical requirements of the Space Station program, and not that a specific piece of hardware meets a new operating condition. Once final requirements are fixed, the technology developed during this program can be used as a basis for producing a piece of flight hardware.

The initial contract required the delivery of all hardware developed during the project. However, the need to deliver a working prototype for long-term endurance testing is outside this requirement. Since the primary objective of this project was not hardware development, the first prototype was developed for performance testing and verification only. Prior to any long-term endurance testing with this unit, it should be upgraded based on results of this initial project.

This final report documents the results for the entire on-orbit compressor technology program. The following chapters summarize the major project phases of conceptual design, detail final prototype design, and prototype verification program, and analysis of results. Detailed phase reports for each of these phases are included in the appendices of the report.

2.0 CONCEPTUAL DESIGNS

The conceptual design layout is illustrated in Figure 1. This design places emphasis on simplicity and a minimum number of components. The requirement for commonality is met by a discrete number of cylinder bores and stroke combinations. The cylinder sizes required to satisfy the range of requirements for this program are quite small in displacement, and therefore, warrant unique design concepts to satisfy the range of flow, gas properties, and pressure ratio requirements. High performance and good efficiency are required for these very small, multi-stage compressors. Therefore, we must pay special attention to factors which detract from efficiency, while keeping in mind durability, size, weight and power. The basic efficiency issues for these small cylinders are mechanical friction, valve and piston seal leakage, and heating effects.

The basic design concept is the employment of an eccentric crankshaft with anti-friction bearing driven pistons and piston return springs. Self-lubricated guide and seal rings are used to center the piston in the cylinder and maintain gas pressure. The driving means is a variable speed electric motor and check valves are used for gas suction and discharge. Active cooling is used at the cylinder walls and at the discharge accumulators for each stage.

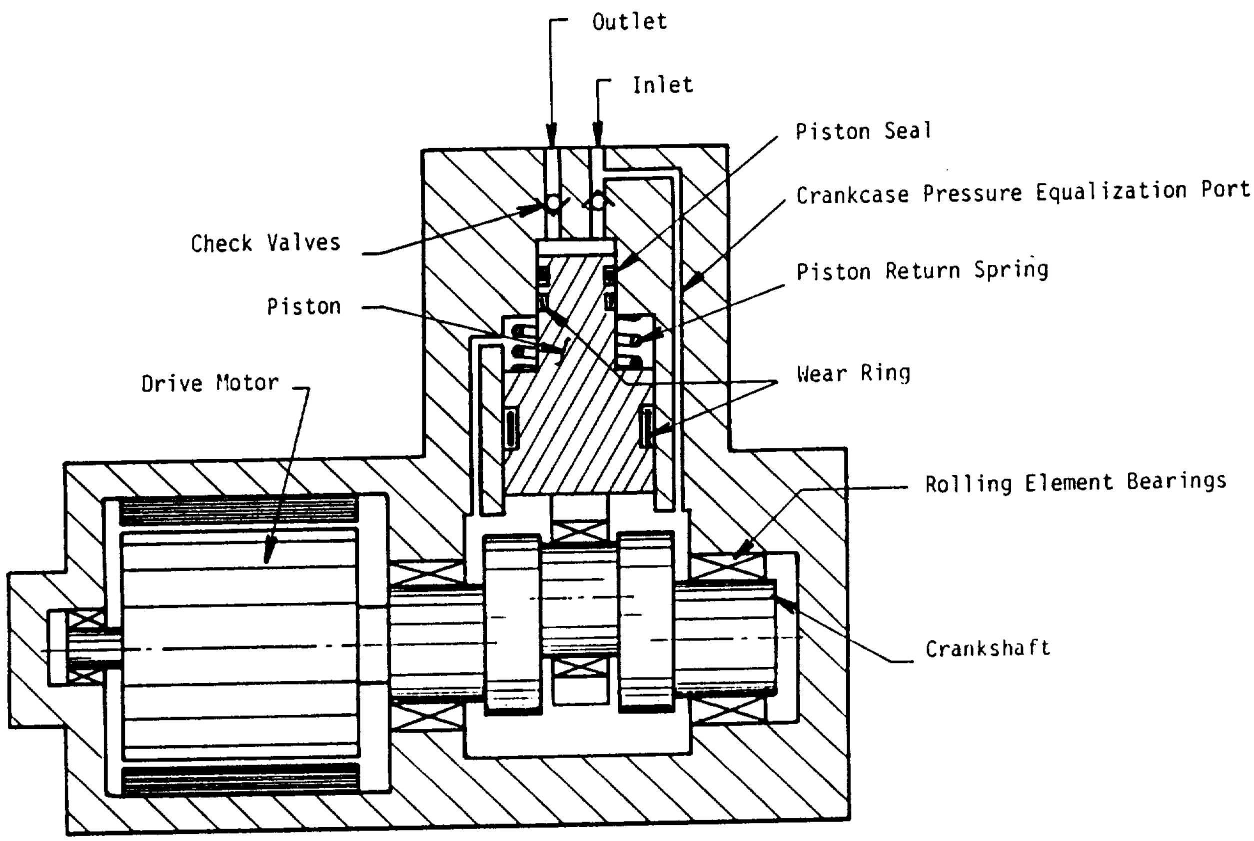

Figure 1. Preliminary Compressor Concept

The compressor requirements for the seven applications cover a wide range of gas properties, pressure, temperature, flow, and duty cycle. The fluid systems of interest are: (1) Space Station Integrated Waste Fluid System (IWFS), (2) Space Station Integrated Nitrogen System (INS), (3) Space Station Propulsion System, and (4) Orbital Space Craft Refueling System (OSCRS). The specific applications are as follows:

Type I:	Space Station IWFS - Reducing Gas Mixture Storage/Disposal
Type II:	Space Station IWFS - Oxidizing Gas Mixture Storage/Disposal
Type III:	Space Station INS - Nitrogen Gas Storage/Resupply
Type IV:	Space Station Propulsion System - Hydrogen Gas Storage/Resupply
Type V:	Space Station Propulsion System - Oxygen Gas Storage/Resupply
Type VI:	OSCRS - Pressurant Gas Resupply
Type VII:	OSCRS - Ullage Gas Recompression

The detailed operation requirements are summarized in Table I. Significant problems and challenges are presented by these requirements, and no existing hardware is currently available to meet the specifications. As an example, the Type I application requires compression of a reducing gas mixture consisting largely of hydrogen, carbon dioxide, and nitrogen from 0.12 MPa (18 psia) up to 8.3 MPa (1200 psia), a total pressure ratio of 67:1, on a continuous basis, with a 10,000 hour operating life goal. Employing a conventional approach, this would be accomplished by a reciprocating compressor using three or four stages driven at relatively high speeds, resulting in very small or miniature cylinders. An early selection for the fourth stage for the Type I application was a cylinder with a 0.0095 meter (0.375 inch) bore, a 0.0022 meter (0.086 inch) stroke driven at up to 3300 RPM. The problems posed by such miniature cylinders, valves, and other compressor components were apparent early on, but developments during this project have allowed a reversal toward more acceptably sized cylinders (larger) driven at low speeds.

The Type VII application presents a unique challenge since the compression of helium saturated with hydrazene, having a maximum safety limit of 71.1°C (160°F), is required from an initial pressure of 2.76 MPa (400 psia) up to 31 MPa (4500 psia). The problem here, based on conventional compression technology, is to achieve this total pressure ratio (11.25:1) employing a reasonable number of stages without ever exceeding 71.1°C (160°F) at any point of the compression cycle. Based on the starting temperature of 51.7°C (125°F), and assuming isentropic compression with no precooling, the first stage

TABLE I. COMPRESSOR APPLICATION REQUIREMENTS

Application	I	II	III	IV	V	VI	VII
System	SS:IWGS-1	SS:IWGS-2	SS:INS	SS:PROP-1	SS:PROP-2	OSCRS-1	OSCRS-2
Fluids	H2	Waste Gas	N2	H2	O2	N2	He
Contaminants	None	Trace Mixture	None	Water Vapor	Water Vapor	None	Propellant Vapor
Pressure: MPa (psia)							
Inlet	0.03 to 0.20 (20 to 30)	0.07 to 0.20 (10 to 30)	3.8 to 4.5 (550 to 650)	1.04 to 1.55 (150 to 225)	1.04 to 1.55 (150 to 225)	27.6 to 2.8 (4000 to 400)	1.7 (250)
Outlet	6.9 to 8.27 (1000 to 1200)	6.9 to 8.27 (1000 to 1200)	40.0 to 42.1 (5900 to 6100)	2.1 to 20.7 (300 to 3000)	2.1 to 20.7 (300 to 3000)	41.4 (6000)	2.8 to 31 (400 to 4500)
Temp: °C (°F) Inlet							
Maximum	32.2 (90)	32.2 (90)	21 (70)	65.6 (150)	65.6 (150)	51.7 (125)	51.7 (125)
Minimum	15.5 (60)	15.5 (60)	-95.6 (-140)	37.8 (100)	37.8 (100)	4.4 (40)	4.4 (40)
Flow Rate: Kg/hr(LBm/hr)							
Nominal	0.005 (0.012)	0.11 (0.25)	4.90 (10.8)	0.078 (0.167)	0.62 (1.36)	400 lbm/24 hrs.	Power Limited
Maximum	0.014 (0.030)	0.500 (1.1)	7.67 (16.9)	0.14 (0.31)	1.59 (3.5)		
Operating Life: hr	10000	10000	10000	25000	25000	1000	1000
Duty Cycle	Continuous	Continuous	30 hrs/90 days	On: 54 min. Off: 36 min.	On: 54 min. Off: 36 min.	24 hrs. Continuous Every 3 months	24 hrs. Continuous Every 3 months
Power: KW Peak	1	1	1	1	1	1	1
Line Size: Meters (inch)							
Inlet	0.01 (3/8)	0.01 (3/8)	0.01 (3/8)	0.01 (3/8)	0.01 (3/8)	0.01 (3/8)	0.01 (3/8)
Outlet	0.01 (3/8)	0.01 (3/8)	0.01 (3/8)	0.0064 (1/4)	0.0064 (1/4)	0.01 (3/8)	0.01 (3/8)

pressure ratio could not exceed 1.156:1, hence, many stages would be required. The thermal problem for this application is further compounded by the lack of a liquid cooled heat sink (ammonia buss duct) which is available for most other applications.

The compressor conceptual design is based on detailed performance modeling and breadboard testing (summarized in the Prototype Design Final Report included in Appendix B) to verify the model. The comprehensive digital time domain model includes cycle thermodynamics, real gas properties, in cylinder heat transfer, valve dynamics, ring leaks, piston friction, attached piping acoustic effects, and piston inertial effects. Besides predicting the compressor performance, the simulation model also provides prediction of several mechanical components such as: actuator bearing B-10 life, seal ring wear, actuator interface contact stress/life, valve stress/life, return spring stress/life, and piston loading.

Table II shows the basic conceptual design and performance information for the compressors for all seven applications which were outlined in Table I. The top section of Table II shows design data, including number of stages, bore, stroke, and cylinder clearance factor (CF). The full range of compression required by these seven applications is covered by three distinct cylinder sizes, or:

- 1 inch Bore x 0.8 inch Stroke
- 0.875 inch Bore x 0.48 inch Stroke
- 0.500 inch Bore x 0.25 inch Stroke

The compressor for the Type II application, which was carried to the prototype hardware stage as part of this project, consists of two stages. The first stage employs two identical cylinders with a bore of 0.875 inches and a stroke of 0.48 inches. The second stage has a single cylinder with a bore and stroke of 0.500 and 0.250 inches, respectively.

All other applications, with the exception of Type III, employ two stages with single cylinders per stage, each stage cylinder differing in bore and stroke. The Type I application employs identical cylinders for first and second stages (0.500 inch bore and 0.250 inch stroke) with proper loading accomplished by the selection of interstage pressure.

The second two sections of Table II show predicted performance data for each application. The suction, interstage, and discharge pressures are denoted by Ps, Pis, and Pd. Effective suction temperature at the cylinder intake valve (not the gas temperature at the compressor unit) is denoted by Ts.

TABLE II. COMPRESSOR CONFIGURATION AND PREDICTED PERFORMANCE

Application	I	II	III	IV	V	VI	VII
Number of Stages	2	2*	*1*	2	2	2	2
S1 - Bore mm (inches)	12.7 (.500)	22.22 (0.875)	22.22 (0.875)	22.22 (0.875)	22.22 (0.875)	25.4 (1.000)	22.22 (.875)
S1 - Stroke " "	6.35 (.250)	12.2 (0.480)	12.2 (0.480)	12.2 (0.480)	12.2 (0.480)	20.3 (0.800)	12.2 (.480)
S2 - Bore " "	12.7 (.500)	12.7 (0.500)	-	12.7 (0.500)	12.7 (0.500)	22.22 (0.875)	12.7 (.500)
S2 - Stroke " "	6.35 (.250)	6.35 (0.250)	-	6.35 (0.250)	6.35 (0.250)	12.2 (0.480)	6.35 (.250)
S1 - CF (%)	10	12	6	6	6	15	10
S2 - CF (%)	10	12	-	9	10	30	10
Ps - MPa (psia)	0.12 (18)	0.17 (25)	3.8 (550)	0.69 (100)	0.69 (100)	2.8 (400)	1.7 (250)
Pis - MPa (psia)	.59 (85)	1.64 (237)	-	4.1 (600)	4.1 (600)	10.4 (1500)	8.3 (1200)
Pd - MPa (psia)	8.27 (1200)	5.6 (815)	42.1 (6100)	20.7 (3000)	20.7 (3000)	41.4 (6000)	31.0 (4500)
Ts - °C (°F)	21 (70)	21 (70)	4.4 (40)	49 (120)	10 (50)	10 (50)	4.4 (40)
Speed - rpm	500	650	500	1250	1000	550	450
Flow Rate Kg/hr (lbm/hr)	.015 (.034)	.17 (0.37)**	8.4 (18.5)	.15 (0.33)	1.3 (2.8)	7.5 (16.6)	.23 (.51)

* The Type II compressor employs two first stage cylinders and one second stage cylinder with the dimensions shown. All other compressors employ one cylinder per stage.

** Based on observed first stage conditions.

Predicted flow rate is shown in the bottom portion of Table II for the conceptual design rotational speeds indicated. In general, each flow rate equals or slightly exceeds the values shown in Table I.

3.0 DETAIL PROTOTYPE DESIGN

The Type II prototype compressor developed under this project is intended for compressing an oxidizing gas mixture for storage/disposal. The waste gas comes from experiment modules and is discharged into a storage vessel initially at a minimum pressure of 100 psia until the vessel is pumped up to the upper design pressure of 1000 psi. Some of the general design specifications for compressor performance, operating environment, and working fluid are given below.

Compressor Performance Requirements:	
Nominal Fluid Flow Rate	0.11 Kg/hr (0.25 LBM/hr)
Maximum Fluid Flow Rate	0.50 Kg/hr (1.10 LBM/hr)
Inlet Pressure	0.07 - 0.20 MPa (10 - 30 psia)
Discharge Pressure	0.69 - 6.90 MPa (100 - 1000 psia)
Maximum Discharge Pressure	8.28 MPa (1200 psia)
Inlet Fluid Temperature	15.5 - 32.2°C (60 - 90°F)
Duty Cycle	Continuous Operation
Operating Life	10,000 hr
Operating Environment:	
Weight Limitation	36.3 Kg (80 LBM)
Size Limitation	1.5 cubic feet
Power Limitation	1.0 KW Peak
Fluid Mixture:	
Nitrogen	60.0%
Argon	19.3%
Oxygen	3.0%
Air	11.3%
Carbon Dioxide	1.9%
Krypton	1.8%
Xenon	0.6%
Helium	0.2%
Trace Reductants	<0.1%

Trace Contaminants	<1.8%
Inlet Dew Point	-30°F

The basic design, shown in Figure 2, is a 3-cylinder, two stage reciprocating piston type compressor with pressure actuated check valves. The pistons are follower actuated by eccentrically mounted anti-friction bearings. The piston is held in contact with the actuator with a preloaded spring. The following list presents the prototype compressor design parameters:

	First Stage		Second Stage
Number of Cylinders	2		1
Cylinder Bore (inches)	0.875		0.500
Compressor Nominal Speed (RPM)	650-1200		650-1200
Piston Displacement (cu. inches/cylinder)	0.288		0.049
Stroke (inches)	0.48		0.25
Clearance Volume (%)	6		10
Number of Suction Valves	2		1
Diameter of Suction Ports (inches)	0.125		0.094
Number of Discharge Ports	1		1
Diameter of Discharge Ports (inches)	0.125		0.094
Piston Guide Bore (inches)	1.250		1.250
Return Spring Preload (LB)	15		15
Return Spring Rate (LB/inch)	60		60
Actuator Bearing Size	206		106
Crank Main Bearing Size		207	
Motor Peak Rated Torque (oz.-in.)		400	
Motor Power at Rated Peak Torque (watts)		510	
Maximum Continuous Output Power (watts)		560	

This design was based on a number of competing design requirements and represents a reasonable trade-off between performance, reliability and life requirements. The design is simple with few moving parts, and based on component wear and life predictions, is free of sudden catastrophic failure modes. Both compressor stages are driven from a single drive motor and crank assembly resulting in fewer mechanical

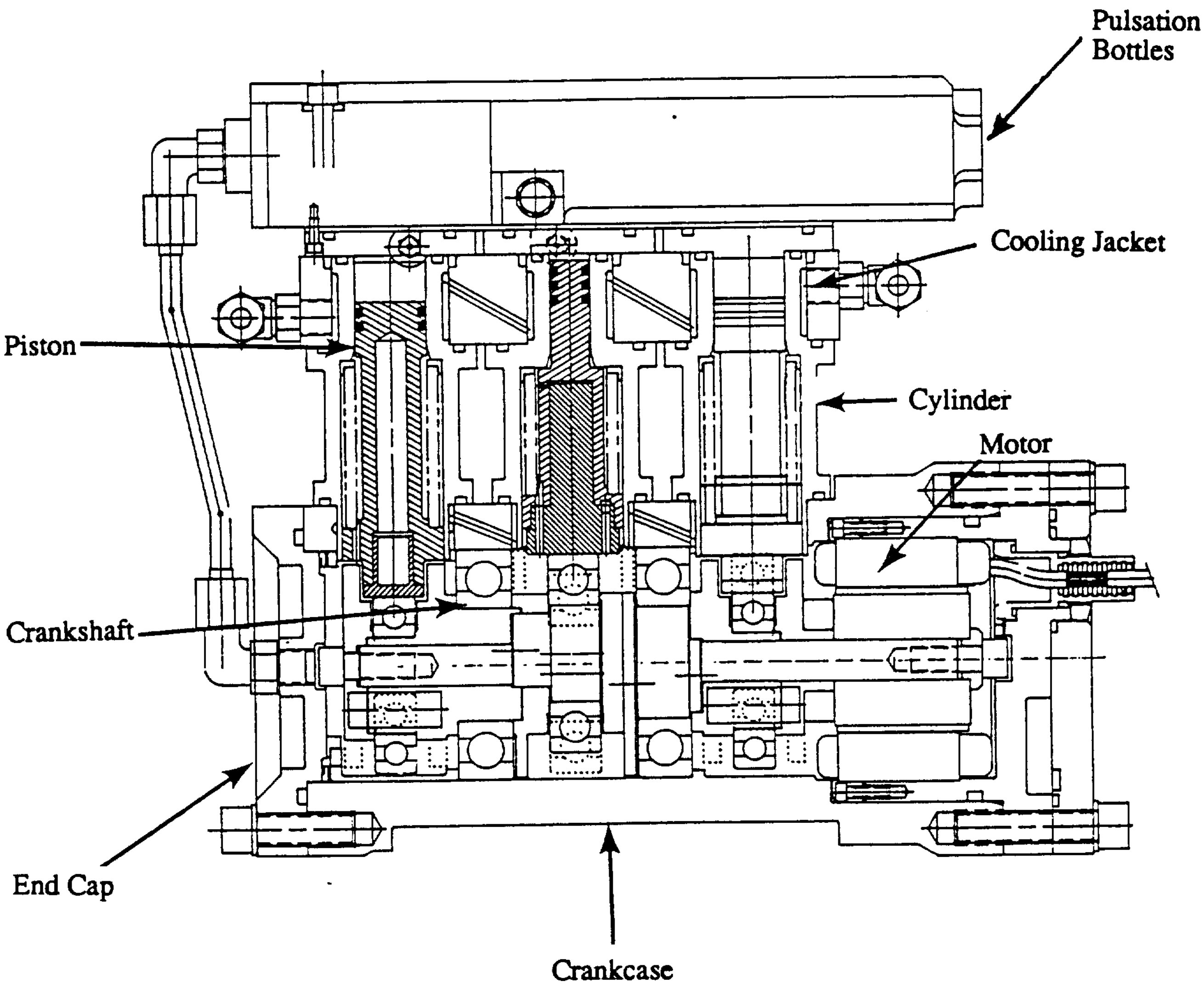

Figure 2. Prototype Compressor Layout

components and lighter weight compared with separate stages. The integration of both stages also simplifies installation since manifolding between the two stages and the pulsation bottles is incorporated into a single head assembly.

The three cylinder design is balanced to eliminate primary and residual secondary shaking forces. While the three cylinder design is somewhat more complicated than other possible designs, the ability to limit shaking forces is very important. The option of an unbalanced compressor with compensating hardware (active or passive devices) was investigated and determined to be unacceptable for a variable speed compressor.

The prototype hardware was fabricated in Southwest Research Institute's Machine Shop and is shown in Figure 3.

4.0 VERIFICATION PROGRAM

The Verification Program provides the methods used to verify that the prototype compressor meets the design requirements outlined in the End Item Specification (EIS). The Verification Program is broken into three sections based upon the verification method employed. The first section describes the verification by development testing. The second section consists of verification by analysis, and the third section consists of verification of assessment.

In order to ensure that the compressor meets the design requirements, some detailed testing of compressor components and the compressor assembly is necessary as indicated in Table III. Development testing is done to substantiate designs, measure performance, and assure the design is suitable for initiation of formal flight hardware development. Since development testing is not intended to provide flight certification, the formal requirements of controlled design, formal certification, formal retest, and flight type hardware are not required.

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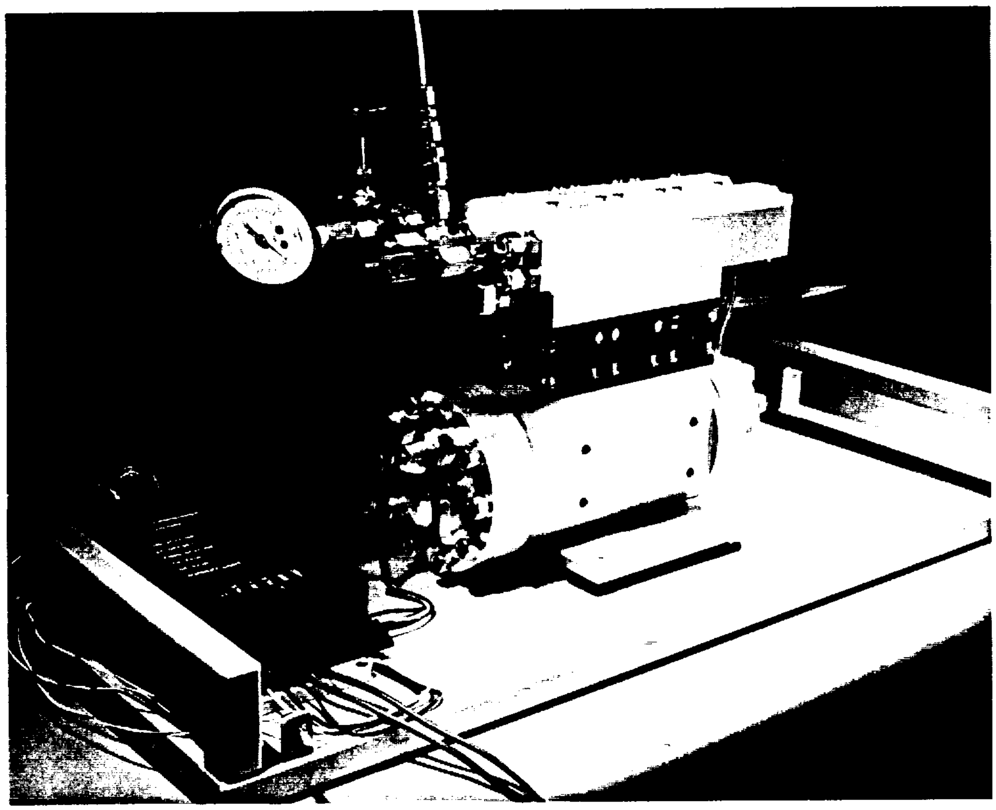


Figure 3. Photograph of Waste Gas Prototype Compressor

Table III. Verification by Test

EIS Section No. & Title		Method of Verification
3.2.2.	Strength	Exempt Except for Proof Pressure - Test
3.2.2.3	Surface Wear	Test & Analysis
3.2.2.5	Weight	Test
3.2.2.6	Envelope	Test
4.2.2.1	Operating Pressures	Test
4.2.2.2	Proof Pressure	Test
4.2.3	Fluid Operating Temperatures	Test
4.2.4	Fluid Flow Rate	Test
4.2.5.1	Operating Life	Test & Analysis
4.2.6	Power Limitations	Test

The results of the surface wear tests performed with the Subassembly Test Article (STA) are shown in Figure 4. The total assembly weight is 30 lbs. and the total envelope size is 0.5 cubic feet. The performance results are presented in Appendix C and are summarized in Figures 5, 6, and 7.

These plots present the raw data in non-dimensional form. The abscissa is a non-dimensional pressure in the form of pressure ratio (discharge pressure divided by suction pressure). The ordinate is a non-dimensional mass flow rate which is, also, the volumetric efficiency. The volumetric efficiency is defined as the mass of gas actually pumped by the compressor, divided by the mass of gas which the compressor could pump, if it handled a volume of gas equal to its piston displacement time compressor speed (in RPM), and if no thermodynamics state changes occurred during the intake stroke (mass flow rate divided by the product of suction density, piston displacement, and RPM).

Table IV lists the EIS items that require verification by analysis. Verification by analysis is primarily used where simulated design conditions cannot be met, test data must be extrapolated beyond the test parameters, and where articles of similar design have been verified to equivalent requirements.

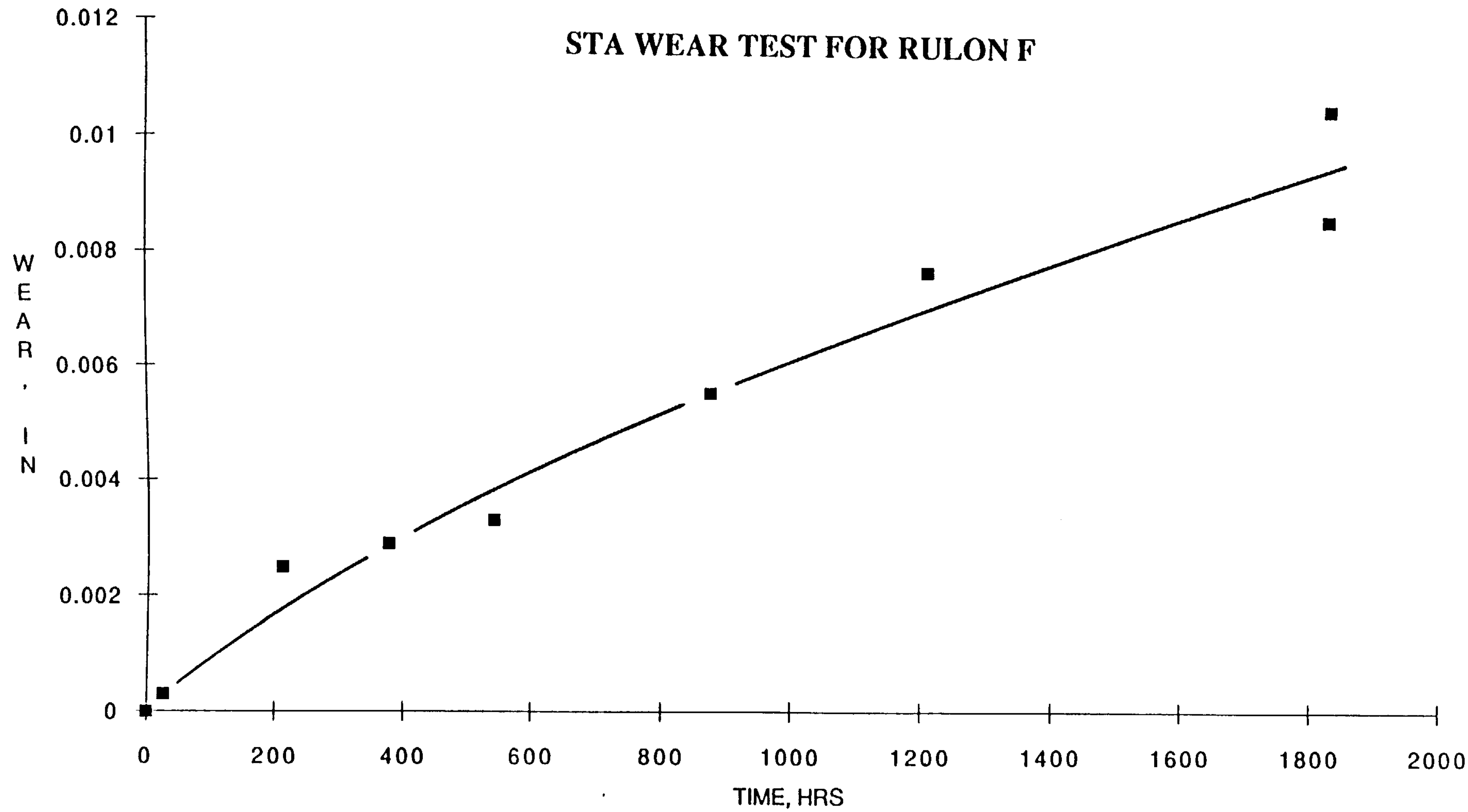


Figure 4. STA Wear Test for Rulon F

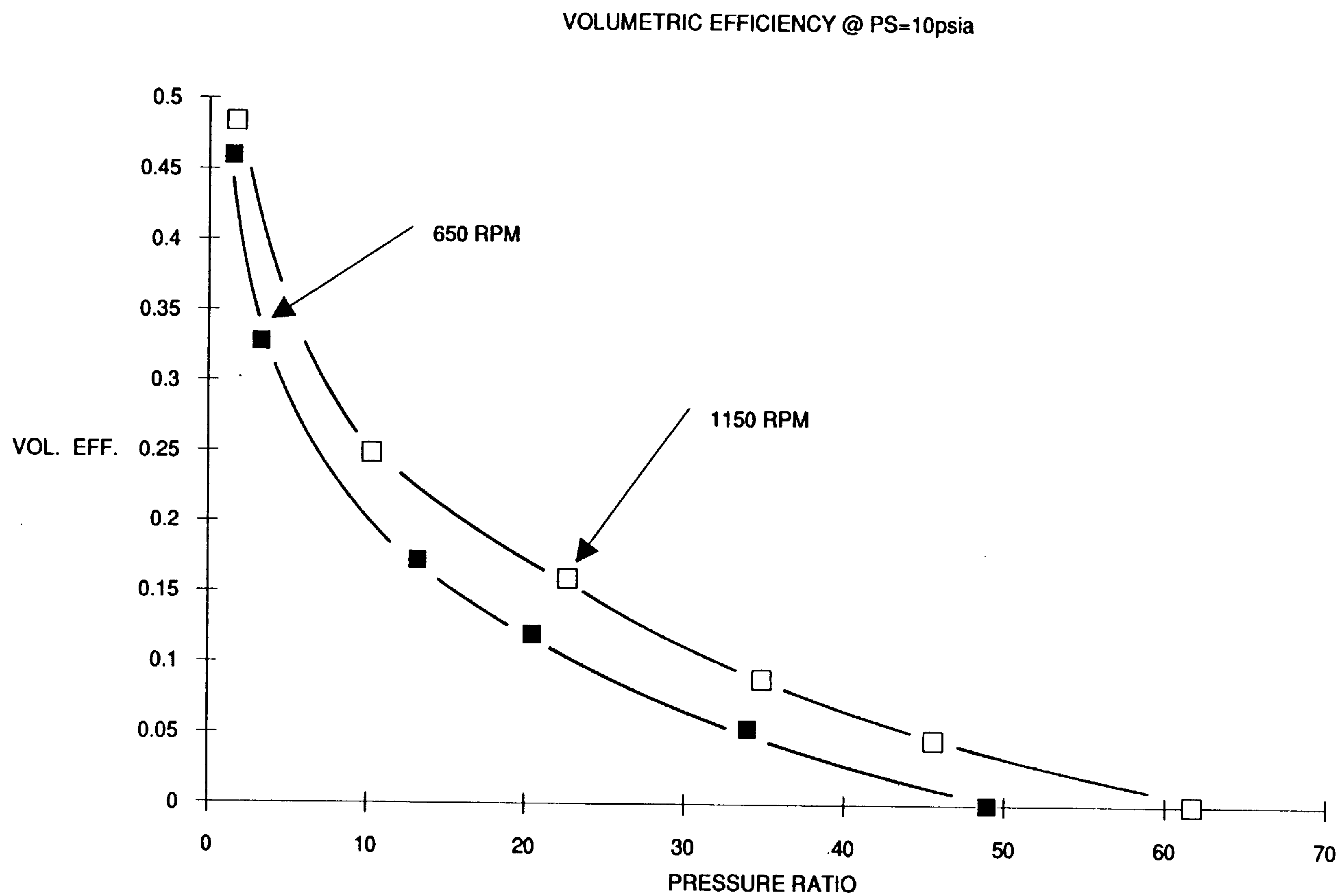


Figure 5. Volumetric Efficiency versus Pressure Ratio at 10 psia Suction Pressure

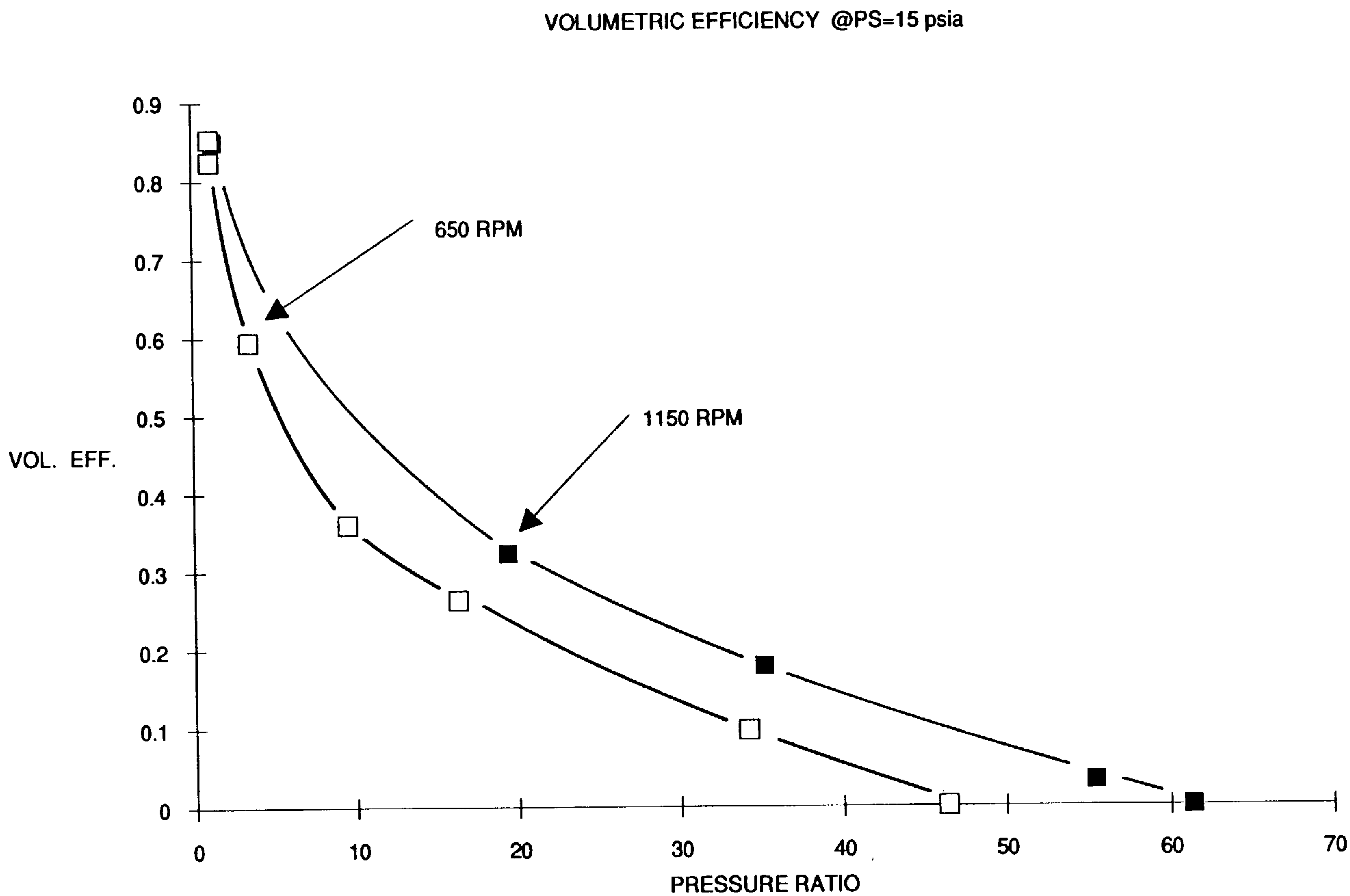


Figure 6. Volumetric Efficiency versus Pressure Ratio at 15 psia Suction Pressure

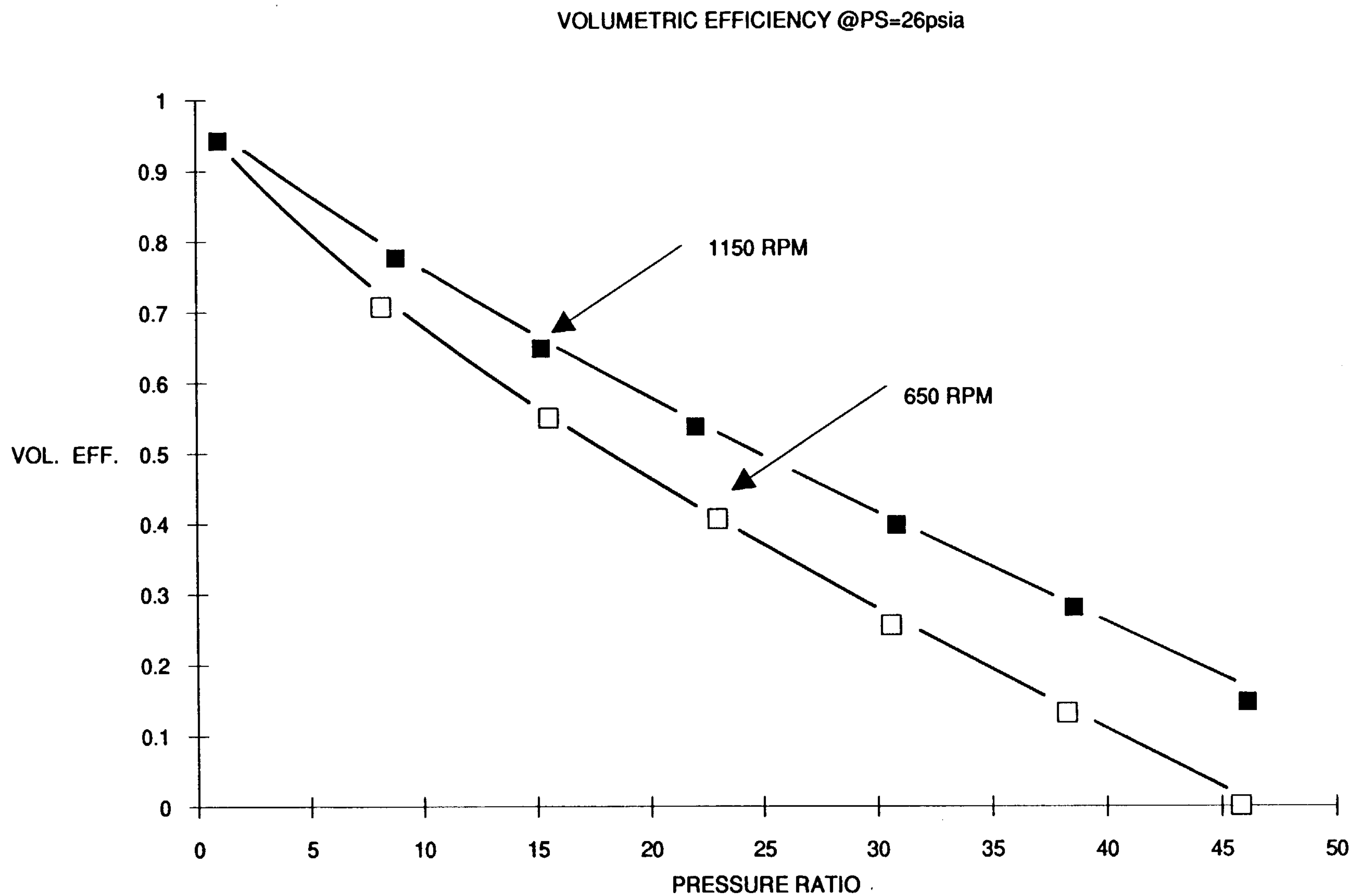


Figure 7. Volumetric Efficiency versus Pressure Ratio at 26 psia Suction Pressure

Table IV. Verification by Analysis		
EIS Section No. & Title		Method of Verification
3.2.2.3	Surface Wear	Analysis & Test
4.2.2.3	Burst Pressure	Analysis
4.2.5.1	Operating Life	Testing & Analysis

The life limiting component for this device is the second stage piston seal ring. The wear data presented in Figure 4 is used to estimate the operating life of the compressor. The seal life analysis is based on the methods given in the ASME Design Manual on PTFE Seals in Reciprocating Compressors[1]. The calculation method is as follows:

Predicted Life ($T_{Pred.}$) in Hours

$$T_{Pred.} = \frac{1}{K_{test}} \left(\frac{R_{limit}N}{P_m V_a}\right)_{projected}$$

$$K_{Test} = \left(\frac{RN}{P_m V_a T}\right)_{test}$$

R_{limit} = % loss of ring thickness

N = number of rings

P_m = $\left(\frac{n}{n-1}\right) P_{suc}\left[(P_{dis}/P_{suc})^{\frac{n-1}{n}} - 1\right]$

n = ratio of gas specific heats

V_a = stroke x RPM/6

A review of the STA seal wear test data indicates that after the initial break-in period, the data shows a constant wear rate. If we use this wear rate and account for the material loss during the break-in period, the above procedure can be used to predict life. Based on assumed nominal pressure conditions, (interstage pressure of 200 psia and

1 American Society of Mechanical Engineers, "Manual of Material Selection, Design and Operating Practices, PTFE Seals in Reciprocating Compressors," ASME, New York, NY 10017, 1975.

discharge pressure of 800 psia), nominal speed (650 RPM), and 50% ring thickness loss, the predicted second stage seal life is 9500 hours. This predicted life, within the uncertainty of the STA data, meets the requirement for minimum compressor life.

The burst pressure analysis indicates that the prototype meets the 2500 psia burst pressure requirement.

Table V lists the EIS items that will be verified by assessment. Verification by assessment requires the careful review and evaluation of design drawings or visual inspections. Verification of EIS requirements by the assessment method is commonly used for verification of surface finishes, tolerances, identification, and items requiring visual inspection. All items meet the assessment requirements.

5.0 ANALYSIS OF RESULTS

The global test data presented above, along with digitized in-cylinder pressure and RPM versus time data, has been analyzed in order to evaluate the overall performance of the prototype compressor. The in-cylinder data, when compared with corresponding simulation results, has provided the best assessment of the detailed performance of the cylinder, including heat transfer, valve and ring leaks, and indicated power. The understanding gained from this comparison has helped in the interpretation of the global test data.

The in-cylinder pressure data sought in the current tests was pressure versus volume curves or "P-V card." Normally, when P-V data is acquired for a reciprocating piston compressor, a direct driven shaft digital encoder is employed to trigger simultaneous storage of various channels of information for a select number of points in one revolution. With the use of an encoder, the effect of compressor speed variation is nullified, and data is obtained at the desired number of evenly spaced angular points.

Due to the integrated drive package, no encoder was able to be installed in the compressor. There was, however, access to an analog instantaneous shaft rotational speed signal in the motor control unit. A variable frequency, external device was used to cause triggering of cylinder pressure and instantaneous RPM data acquisition by a digital computer. Data was acquired over approximately one or more revolutions of the compressor, with initiation of acquisition occurring at an arbitrary and unknown shaft position. Approximately 650 points of data per revolution were acquired.

Table V. Verification by Assessment

EIS Section No. & Title		Method of Verification
3.1.1.2	Mechanical	Assessment
3.1.1.3	Electrical	Assessment
3.1.2.2	Lubrication	Assessment
3.2.3	Safety, Reliability, and Quality Assurance	Assessment
3.2.4.1	Transportation	Assessment
3.2.4.2	Storage in Protected Areas	Assessment
3.2.5	Transportability	Assessment
3.3.1.1	Materials and Processes	Assessment
3.3.1.2	Prohibited Materials	Assessment
3.3.1.3	Fluids	Assessment
3.3.1.4	Material Compatibility	Assessment
3.3.1.5	Lubricants	Assessment
3.3.1.6	Dissimilar Metals	Assessment
3.3.1.7	Platings and Castings	Assessment
3.3.1.8	Protective Treatment	Assessment
3.3.1.11	Non-Destructive Evaluation	Assessment
3.3.1.12	Cleanliness	Assessment
3.3.1.15	Assembly Cleanliness	Assessment
3.3.1.16	Parts Standardization	Assessment
3.3.1.17	Threads and Fasteners	Assessment
3.3.1.18	Locking Threaded Parts	Assessment
3.3.1.19	Prohibited Retaining Methods	Assessment
3.3.1.20	Surface Texture	Assessment & Test
3.3.1.21	Dimensioning and Tolerancing	Assessment
4.2.1	Fluid	Assessment
4.2.7	Line Sizes	Assessment

In summary, the above data consisted of the files of corresponding cylinder pressure and instantaneous RPM data starting at unknown shaft position points, containing approximately 650 points per revolution. It was necessary to extract data representing only one revolution, and then convert this data to a usable file of 512 points per revolution representing cylinder pressure and shaft angle, to represent data that would be obtained by the use of a shaft driven encoder. Once this was accomplished, existing digital software was employed to compare test P-V cards with corresponding simulation results.

Data from two cases which represent opposite extremes for which in-cylinder pressure was obtained is shown in Figures 8 and 9. The pressure versus volume data of Figure 8 is for a low pressure ratio and maximum flow, while the data of Figure 9 is for a deadhead pressure case with no flow. Both data sets are for a compressor speed of approximately 650 RPM.

The data in Figure 8, which is an actual test P-V card for the prototype compressor first stage, may in general be distinctively different from an ideal card for the same compressor operating between the same suction and discharge pressures, and without heat transfer or leaks. Figure 10 shows a comparison of the test card of Figure 8 with simulation results for a corresponding ideal compressor cylinder with no leaks or heat transfer. Note the difference between the two cards, particularly along the compression line. Also, the test card shows evidence of a slightly late discharge valve closure which is not indicated on the ideal card.

By varying heat transfer, leak parameters, and introduction of a late discharge valve in the simulation model, it is possible to "force" an approximate agreement between the test and simulation P-V cards. The results in Figure 11 show results of such a "forcefit." While not perfect, the results indicate that the greatest difference between the ideal simulation and test result comparison in Figure 10 is largely a result of heat transfer, with various ring and valve leaks playing an additional part. The effect of heat transfer is generally to enhance performance, while leaks detract from performance.

A similar procedure to the above has been used to evaluate the case of Figure 9. The results were essentially the same as discussed above in that the difference between an ideal simulation and the test card could be reconciled by heat transfer plus minor valve and ring leaks.

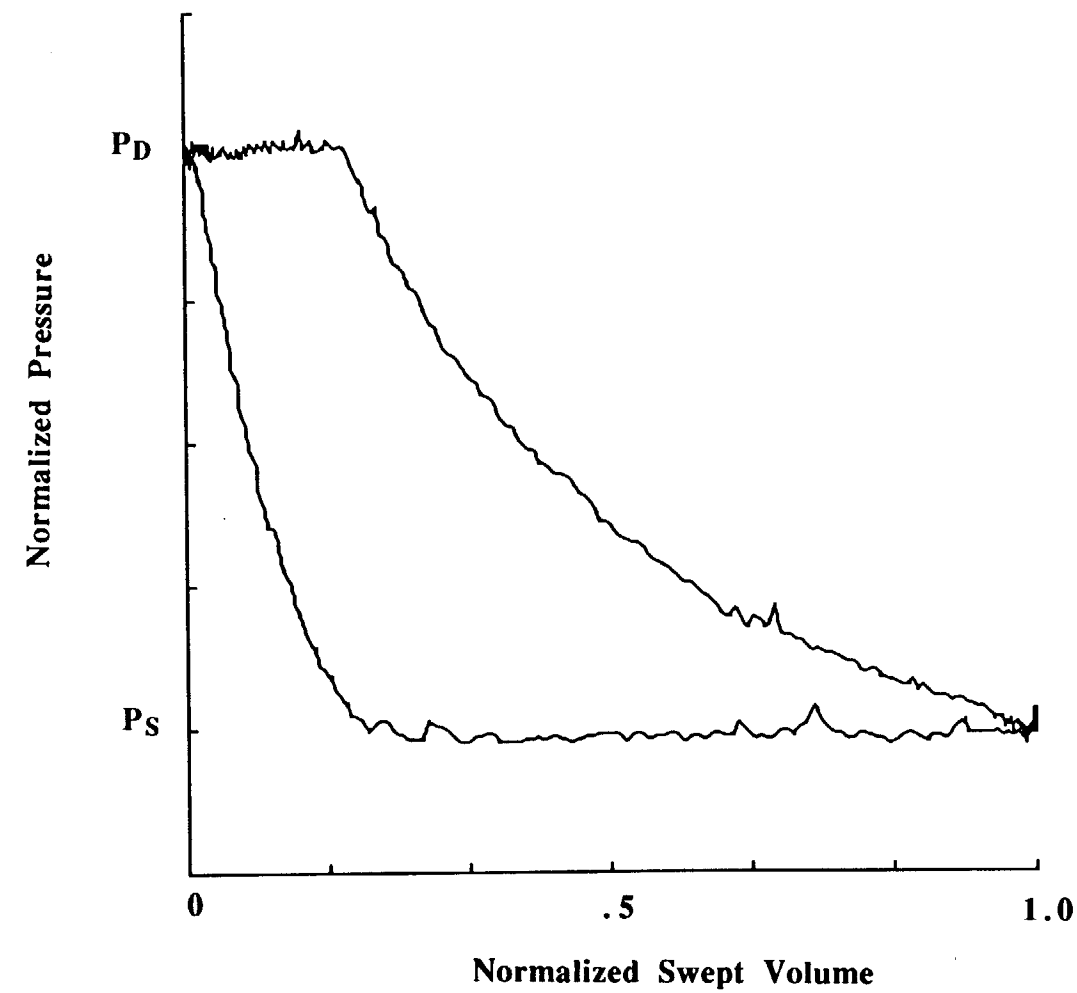


Figure 8. Normalized Pressure Versus Volume for One First Stage Cylinder; P_s = 14.5 psia, P_d = 50 psia, N = 650 RPM

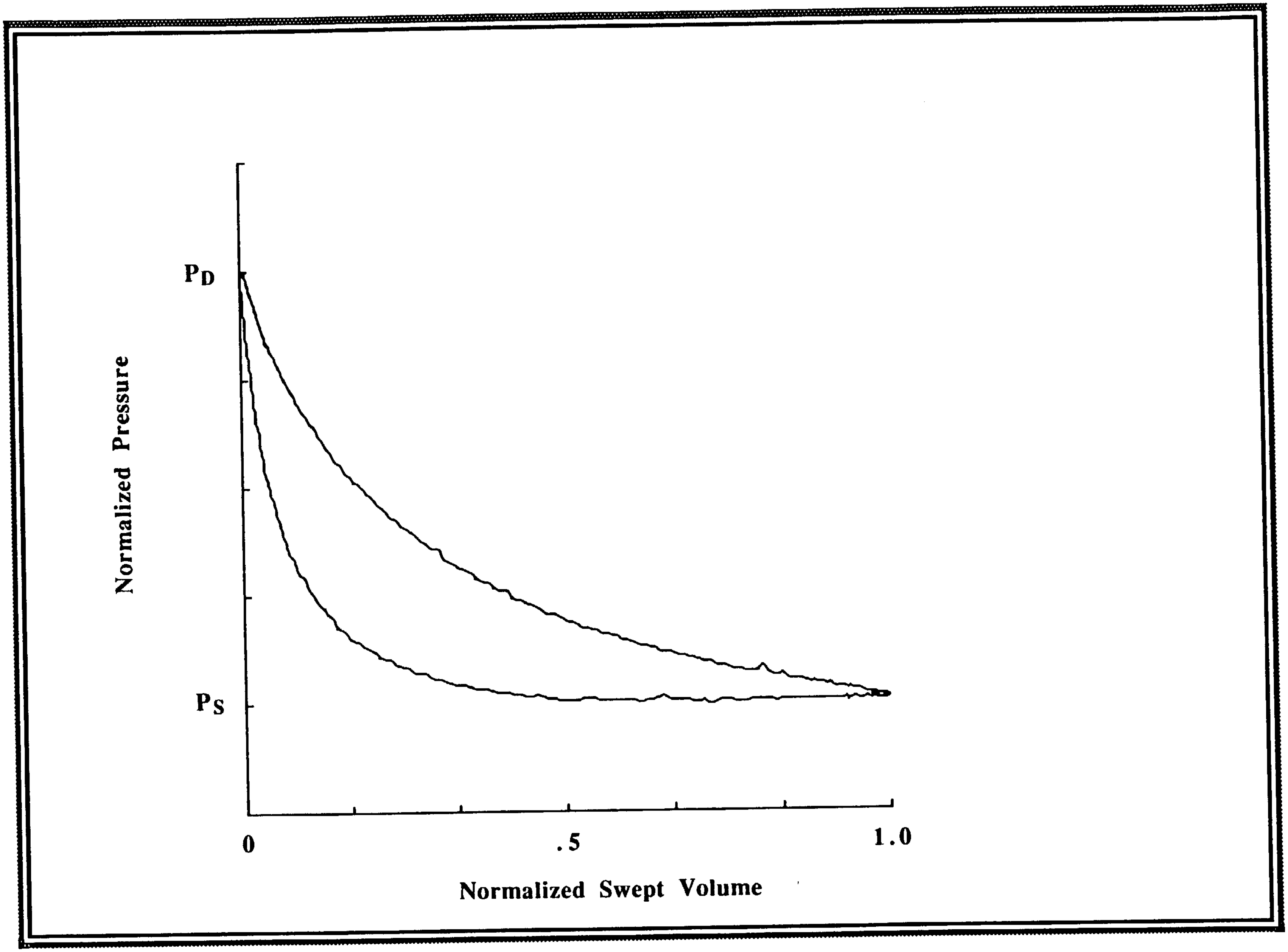


Figure 9. Test Pressure Versus Volume Card for First Stage Cylinder Operating with Zero Net Flow Through Compressor

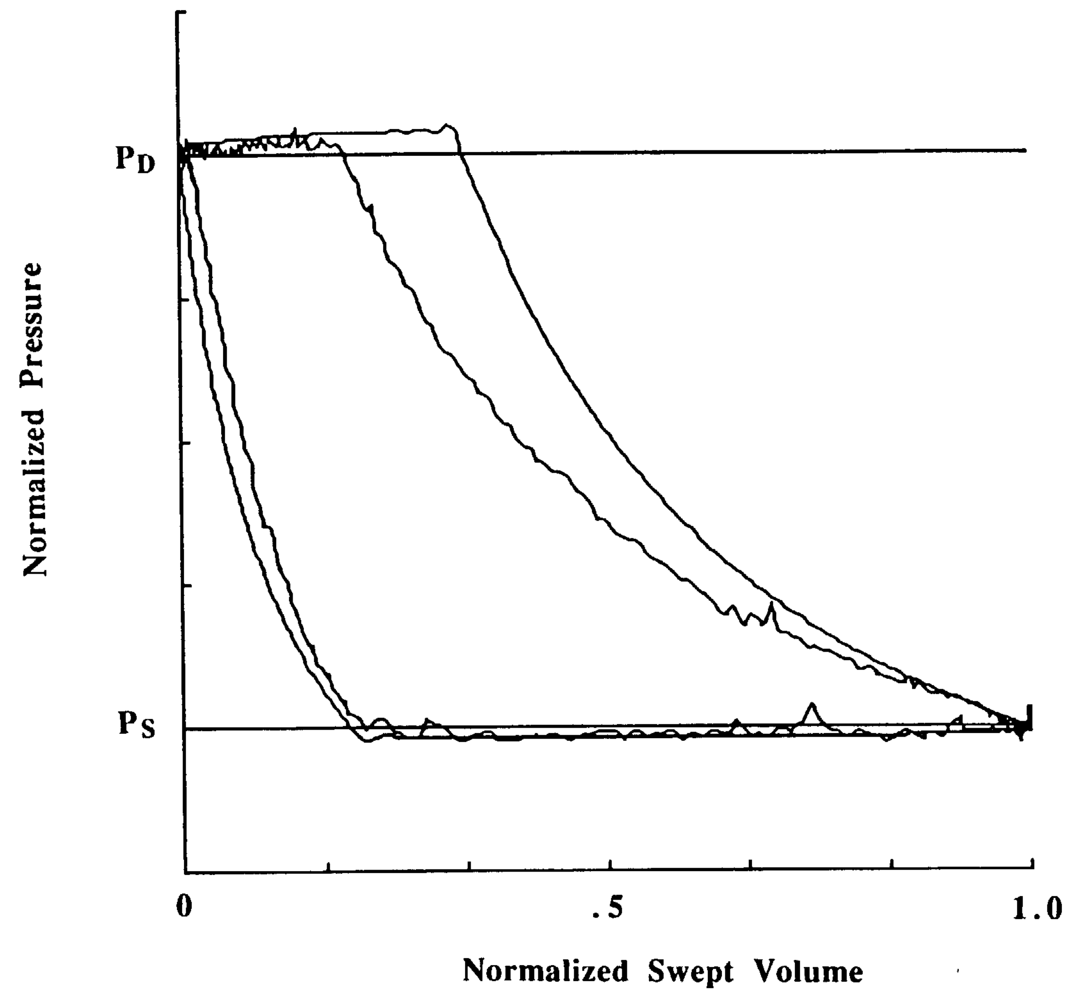


Figure 10. Overlay of Ideal Simulation with No Leaks for Heat Transfer on P-V Card of Figure 8

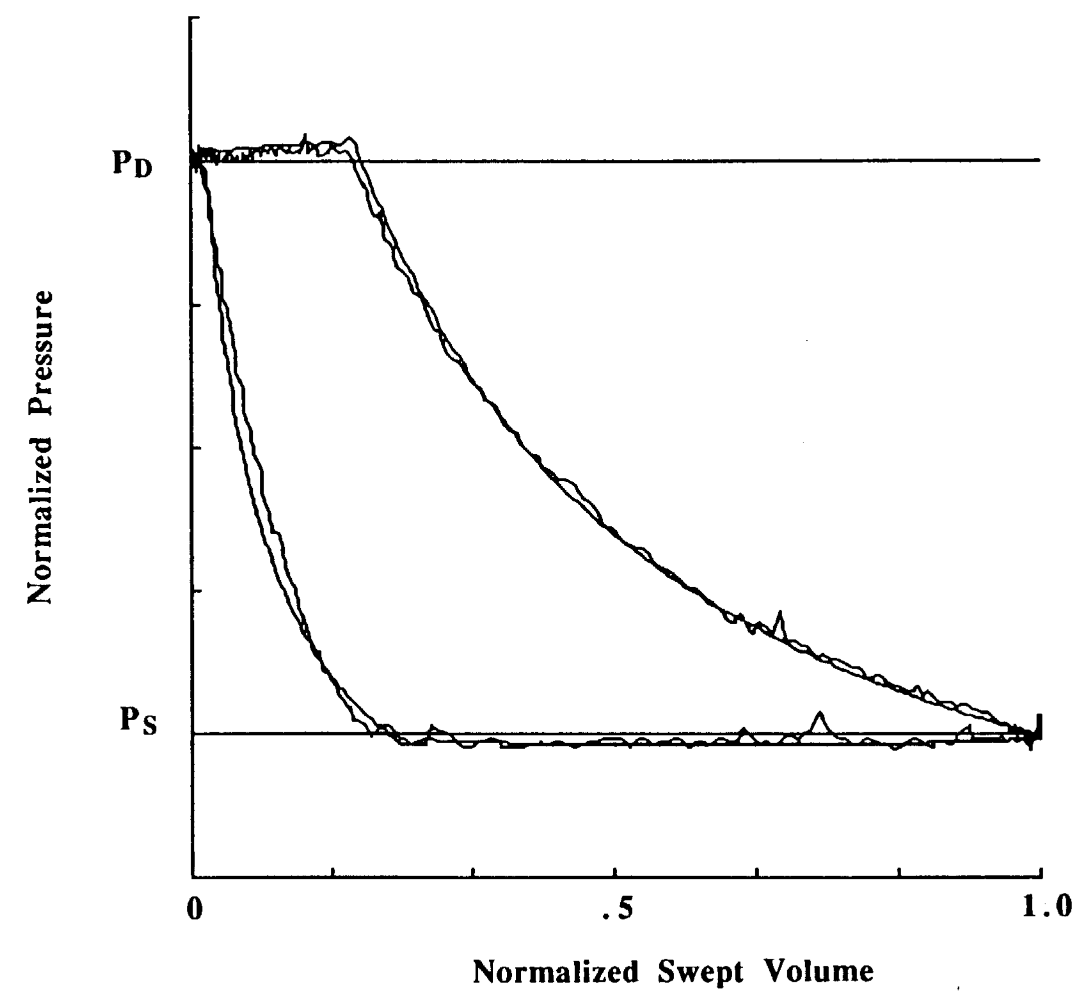


Figure 11. Overlay of Simulation with Heat Transfer, Appropriate Leaks, and Late Discharge Valve on Test P-V Card of Figure 8

The conclusions from the analysis described above and the global data are:

(1) The compressor first stage cylinders are working generally as expected with considerable enhancement from heat transfer, but minor valve and ring leaks detract somewhat from performance. These leaks were to be expected.

(2) The second stage cylinder has a rather sizeable leak around the valves and/or rings. Most likely the leakage is occurring around the rings.

(3) The global test data generally shows trends which agree with the conclusions above, and specifically:

- For a given suction and discharge pressure, the pressure ratio across the first stage is equal to what is expected, while the pressure ratio across the second stage is somewhat lower than expected. This supports the conclusion that there is more leakage on the second stage than the first stage.

- The shut-off deadhead pressure increases as speed increases for the global test data, again reflecting in the leakage occurring in the second stage.

(4) The higher leakage occurring in the second stage cylinder is not inherent in the valve and ring design, but rather in the execution of the miniature size of the rings and valve seats.

6.0 CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

The verification program has documented that the on-orbit compressor prototype meets the requirements of the End Item Specification (EIS) relevant to prototype hardware. The prototype compressor is 3/8 of the allowable weight (30 lbs. versus 80 lbs.), 1/3 of the allowable volume (0.5 cu. ft. versus 1.5 cu. ft.), and 1/2 of the allowable power (500 watts versus 1000 watts). The performance requirements of flow rate, discharge pressure, and suction pressure were independently verified. At a suction pressure of 27 psia and a compressor speed of 650 RPM, the prototype developed a deadhead pressure of 1210 psia, a 0.124 LBM./hr. flow rate at a 1000 psia discharge, a nominal flow rate of 0.24 LBM./hr

at 800 psia discharge and 1.827 LBM/hr. flow rate at zero pressure rise. This performance clearly meets the EIS requirements of an outlet pressure range of 100 to 1000 psia (1200 psia maximum), and a nominal flow rate of 0.25 LBM/hr (1.1 LBM/hr. maximum). The one area of marginal performance is the leakage and life of the second stage piston seals. With the current Space Station interest in significantly longer life than the EIS requirement of 10,000 hours, we recommend that this area be further developed. We specifically recommend that a lubricated seal ring technology be developed for improved sealing and significant life extension.

The overall conclusion is that we have developed compressor technology for on-orbit applications. This technology balances all of the complex design requirements and is provided within a time frame consistent with the support of the Space Station Fluid Systems Development. The verification program has documented that the performance of the prototype waste gas compressor does indeed meet the EIS requirements.

6.2 Recommended Modifications to EIS

In general, the EIS represents a good specification for flight hardware. There are a few sections that can be improved for the specific application of an on-orbit compressor. The sections that should be modified are: Surface Wear (3.2.2.3), Lubricants (3.3.1.5), Performance (4.2.2 & 4.2.4), Proof Pressure (4.2.2.2), and Service Life (4.2.5).

6.2.1 Surface Wear

The current EIS states that, ". . . shall not introduce contaminant into the fluid flow path. . ." Each compressor type has a different application, and the effect of wear particles is different. Specifically, a realistic acceptable number and size of wear particle for the waste gas compressors should be stated.

6.2.2 Lubricants

The current EIS states that ". . . do not introduce contamination by entering the fluid flow path." As indicated above for wear particles, a realistic acceptable level of lubricant transfer downstream for the waste gas compressor should be stated.

6.2.3 Performance

The current EIS provides inlet and outlet pressure ranges and a flow range independent of each other. Since the flow rate is not independent of inlet and outlet pressure, specific combined operating conditions should be stated. As an example, at an

inlet (suction) pressure of 10 psia and an outlet (discharge) pressure of 1000 psia, the fluid flow rate shall be 0.25 lbm/hr. The performance of a compressor is best illustrated in the form of a pressure versus fluid flow rate curve at a given rotational speed. The performance curve can be specified by three points: the pressure at zero flow rate (i.e., deadhead pressure), the flow rate at zero pressure rise (i.e, maximum flow rate with suction pressure equal to discharge pressure), and a nominal flow rate at a nominal pressure. This approach to specifying performance assumes a constant compressor rotational speed. Each rotational speed will have a different performance curve. The control strategy for motor speed is also important. A constant speed compressor greatly simplifies the control system, but a variable speed system provides more flexibility in pressure versus flow rate combinations. Since the waste gas application is to pump up a reservoir from 100 psia to 1000 psia, the important issue is at what flow rate. At a constant speed, the flow rate will be high at 100 psia, i.e, the beginning of the cycle and gradually decline as the vessel pressure approaches 1000 psia. If a constant flow rate is required over the entire range of discharge pressures, then a variable speed is required. This capability will result in a more complex control system and a larger capacity unit. Once the operating conditions are determined, then a more specific performance specification can be written with the above guidelines in mind.

6.2.4 Proof Pressure

The current EIS requires that the entire compressor be subjected to a proof pressure of 1.5 times the maximum discharge pressure for five minutes and designed for a burst pressure of 2.5 times the maximum discharge pressure. Since the case is vented to suction pressure, the requirement that the case withstand this proof pressure results in a significantly heavier case than if the case proof pressure was 1.5 times the highest pressure it would experience (i.e., suction pressure). To be more specific, the design burst pressure for the case is 2.5 times the maximum discharge pressure or 2500 psia. However, this is a hundred times the maximum suction pressure. A more realistic requirement would greatly reduce the weight of the case and total weight of the compressor.

6.2.5 Service Life

The current EIS requires 10,000 operating hours of continuous duty. This life requirement with an unlubricated compressor is severely pushing the state-of-the-art (SOA). As the program proceeded, the objective of maximizing life potentially up to 10 years (876,000 hours) was recommended which is beyond the SOA. Two specific modifications are recommended for the waste gas compressor. The first is to identify a

realistic duty cycle, and the second is to allow a lubricated unit. A realistic duty cycle can result in significant factors of life extension, i.e., if the compressor is realistically only on 1/4 of the time the life can be extended by a factor of 4 for dry seals. For lubricated seals, the life prediction is more complex because wear is not only a factor of operating life but also the number of starts. The life prediction for a lubricated unit, also, has the complexity of the life of the lubricant. However, even with these added complexities in life prediction, the life of lubricated units is significantly longer than unlubricated units, and has the chance of meeting long-term Space Station life goals. The specification should be modified to include operating hours and duty cycle.

APPENDIX A

CONCEPTUAL DESIGN FINAL REPORT FOR ON-ORBIT COMPRESSOR TECHNOLOGY PROGRAM

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

1.0 INTRODUCTION

The objective of this project is the exploration of compressor technology applicable for use by the Space Station Fluid Management System (FMS), Space Station Propulsion System, and related on-orbit fluid transfer systems. The approach is to perform compressor research on a breadboard test article, to utilize the results to develop a conceptual design to handle seven different applications, and to develop a prototype for one of these specific applications. The prototype development will be based on the general conceptual design and consist of a detailed prototype design, fabrication of the prototype, and testing of the prototype. The primary emphasis is to develop basic compressor technology (designs, materials, and manufacturing techniques) in a time frame consistent with the support of the Space Station fluid systems development. Design considerations will include: (1) maximization of service life; (2) commonalty; i.e., interchangeability of common hardware assemblies; (3) ease of maintenance; (4) lightweight; (5) small size; and (6) low power.

The purpose of this Conceptual Design Final Report is to document the conceptual design for seven Space Station applications.

2.0 COMPRESSOR APPLICATIONS AND REQUIREMENTS

The compressor requirements for the seven applications cover a wide range of gas properties, pressure, temperature, flow, and duty cycle. These requirements are summarized in Table I. Significant problems and challenges are presented by these requirements, and no existing hardware is currently available to meet the specifications. As an example, the Type I application requires compression of a reducing gas mixture consisting largely of hydrogen, carbon dioxide, and nitrogen from 0.12 MPa (18 psia) up to 8.3 MPa (1200 psia), a total pressure ratio of 67:1, on a continuous basis, with a 10,000 hour operating life goal. Employing a conventional approach, this would be accomplished by reciprocating compressors using three or four stages driven at relatively high speeds, resulting in very small or miniature cylinders. An early selection for the fourth stage for the Type I application was a cylinder with a 0.0095 meter (0.375 inch) bore, a 0.0022 meter (0.086 inch) stroke driven at up to 3300 rpm. The problems posed by such miniature cylinders, valves, and other compressor components were apparent early on, but developments during this project have allowed a reversal toward more acceptably sized cylinders (larger) driven at low speeds.

TABLE I. COMPRESSOR APPLICATION REQUIREMENTS

Application	I	II	III	IV	V	VI	VII
System	SS:IWGS-1	SS:IWGS-2	SS:INS	SS:PROP-1	SS:PROP-2	OSCRS-1	OSCRS-2
Fluids	H2	Waste Gas	N2	H2	O2	N2	He
Contaminants	None	Trace Mixture	None	Water Vapor	Water Vapor	None	Propellant Vapor
Pressure: MPa (psia)							
Inlet	0.03 to 0.20 (20 to 30)	0.07 to 0.20 (10 to 30)	3.8 to 4.5 (550 to 650)	1.04 to 1.55 (150 to 225)	1.04 to 1.55 (150 to 225)	27.6 to 2.8 (4000 to 400)	1.7 (250)
Outlet	6.9 to 8.27 (1000 to 1200)	6.9 to 8.27 (1000 to 1200)	40.0 to 42.1 (5900 to 6100)	2.1 to 20.7 (300 to 3000)	2.1 to 20.7 (300 to 3000)	41.4 (6000)	2.8 to 31 (400 to 4500)
Temp: °C (°F)							
Inlet							
Maximum	32.2 (90)	32.2 (90)	21 (70)	65.6 (150)	65.6 (150)	51.7 (125)	51.7 (125)
Minimum	15.5 (60)	15.5 (60)	-95.6 (-140)	37.8 (100)	37.8 (100)	4.4 (40)	4.4 (40)
Flow Rate: Kg/hr(LBm/hr)							
Nominal	0.005 (0.012)	0.11 (0.25)	4.90 (10.8)	0.078 (0.167)	0.62 (1.36)	400 lbm/24 hrs.	Power Limited
Maximum	0.014 (0.030)	0.500 (1.1)	7.67 (16.9)	0.14 (0.31)	1.59 (3.5)		
Operating Life: hr [8]	10000	10000	10000	25000	25000	1000	1000
Duty Cycle	Continuous	Continuous	30 hrs/90 days [13]	On: 54 min. Off: 36 min.	On: 54 min. Off: 36 min.	24 hrs. Continuous Every 3 months	24 hrs. Continuous Every 3 months
Power: KW Peak [9]	1	1	1	1	1	1	1
Line Size: Meters (inch)							
Inlet	0.01 (3/8)	0.01 (3/8)	0.01 (3/8)	0.01 (3/8)	0.01 (3/8)	0.01 (3/8)	0.01 (3/8)
Outlet	0.01 (3/8)	0.01 (3/8)	0.01 (3/8)	0.0064 (1/4)	0.0064 (1/4)	0.01 (3/8)	0.01 (3/8)

The Type VII application presents a unique challenge since the compression of helium saturated with hydrazene, having a maximum safety limit of 71.1°C (160°F), is required from an initial pressure of 2.76 MPa (400 psia) up to 31 MPa (4500 psia). The problem here, based on conventional compression technology, is to achieve this total pressure ratio (11.25:1) employing a reasonable number of stages without ever exceeding 71.1°C (160°F) at any point of the compression cycle. Based on the starting temperature of 51.7°C (125°F), and assuming isentropic compression with no precooling, the first stage pressure ratio could not exceed 1.156:1, hence, many stages would be required. The thermal problem for this application is further compounded by the lack of a liquid cooled heat sink (ammonia buss duct) which is available for most other applications.

3.0 SUMMARY OF TECHNICAL CHALLENGES

A major challenge of this project has been to provide enhanced and controlled heat transfer through the compressor cycle in order to: (1) achieve high efficiency, (2) limit cycle and discharge temperatures, and (3) reduce the required number of compression stages. Control of cycle temperature is also important with respect to component life, control of liquids and, particularly, to the Type VII application, to stay below the hydrazene safety temperature limit.

Other technical challenges have included: (1) design of long life and dynamically stable valves, (2) efficient control of pulsations, (3) meeting the power, weight, and envelope requirements, and (4) achieve the life and maintainability goals. All of this must be accomplished with component commonality as an important consideration.

4.0 FACTORS AFFECTING COMPRESSOR PERFORMANCE

4.1 Heat Transfer

Heating related phenomena can affect both the volumetric and thermal efficiency of a reciprocating compressor. Two separate mechanisms are involved: one is direct convective transfer of thermal energy to and from the gas and compressor mechanical structure; and the other is an indirect heating mechanism related to the throttling which occurs across the compressor suction valves. In general, improperly controlled heat

transfer reduces flow, thermal efficiency, and often increases the gas discharge temperature. With conventional commercial compressors, flow and efficiency reductions easily exceed 10 to 15 percent from thermal causes alone.

4.2 Compressor Valves

Valves are the single most failure-prone component on reciprocating compressors used, for example, on gas transmission service, where they must operate continuously and, therefore, build up operational stress cycles quickly. Many different types and configurations of compressor valves have been and are being used, including reed, single and multiple poppet, simple plate types, multiple concentric ring, and complex multiple-degree-of-freedom plate types.

Compressor valves fail for a number of reasons. By failing, they no longer perform the flow check function and thus allow leakage, and this is generally a result of breakage. Another cause of failure is thermally-induced distortion. Some leakage may be acceptable; too much is not.

4.3 Pulsation Effects

Another potential problem area with reciprocating compressors is the possible interaction between valve dynamics and pulsations which may exist external to the compressor valves. It has been demonstrated in laboratory tests and in the field that pulsations can induce early or late valve closure, and also valve cocking effects which result in premature valve failures.

5.0 PROTOTYPE COMPRESSOR CONCEPT

The compressor conceptual design is based on detailed performance modeling and breadboard testing (summarized in the Prototype Design Final Report) to verify the model. The comprehensive digital time domain model includes cycle thermodynamics, real gas properties, in cylinder heat transfer, valve dynamics, ring leaks, piston friction, attached piping acoustic effects, and piston inertial effects. Besides predicting the compressor performance, the simulation model also provides prediction of several mechanical components such as: actuator bearing B-10 life, seal ring wear, actuator interface contact stress/life, valve stress/life, return spring stress/life, and piston loading.

The Type II Prototype Design, shown in Figure 1, is a 3-cylinder, two stage reciprocating piston type compressor with pressure actuated check valves. The pistons are follower actuated by eccentrically mounted anti-friction bearings. The piston is held in contact with the actuator with a preloaded spring. The following list presents the prototype compressor design parameters:

	First Stage	Second Stage
Number of Cylinders	2	1
Cylinder Bore (inches)	0.875	0.500
Compressor Nominal Speed (RPM)	650-1000	650-1000
Piston Displacement (cu. inches/cylinder)	0.288	0.049
Stroke (inches)	0.48	0.25
Clearance Volume (%)	6	10
Number of Suction Valves	2	1
Diameter of Suction Ports (inches)	0.125	0.094
Number of Discharge Ports	1	1
Diameter of Discharge Ports (inches)	0.125	0.094
Piston Guide Bore (inches)	1.250	1.250
Return Spring Preload (LB)	15	15
Return Spring Rate (LB/inch)	60	60
Actuator Bearing Size	206	106
Crank Main Bearing Size	207	
Motor Peak Rate Torque (oz.-in.)	400	
Motor Power at Rated Peak Torque (watts)	510	
Maximum Continuous Output Power (watts)	560	

The above outlined design was based on a number of competing design requirements and represents a reasonable trade-off between performance, reliability and life requirements. The design is simple with few moving parts, and based on component wear and life predictions, is free of sudden catastrophic failure modes. Both compressor stages are driven from a single drive motor and crank assembly resulting in fewer mechanical components and lighter weight compared with separate stages. The integration of both stages also simplifies installation since manifolding between the two stages and the pulsation bottles is incorporated into a single head assembly.

The three cylinder design is balanced to eliminate primary and residual secondary shaking forces. While the three cylinder design is somewhat more complicated than other possible designs, the ability to limit shaking forces is very important. The option of an unbalanced compressor with compensating hardware (active or passive devices) was investigated and determined to be unacceptable for a variable speed compressor.

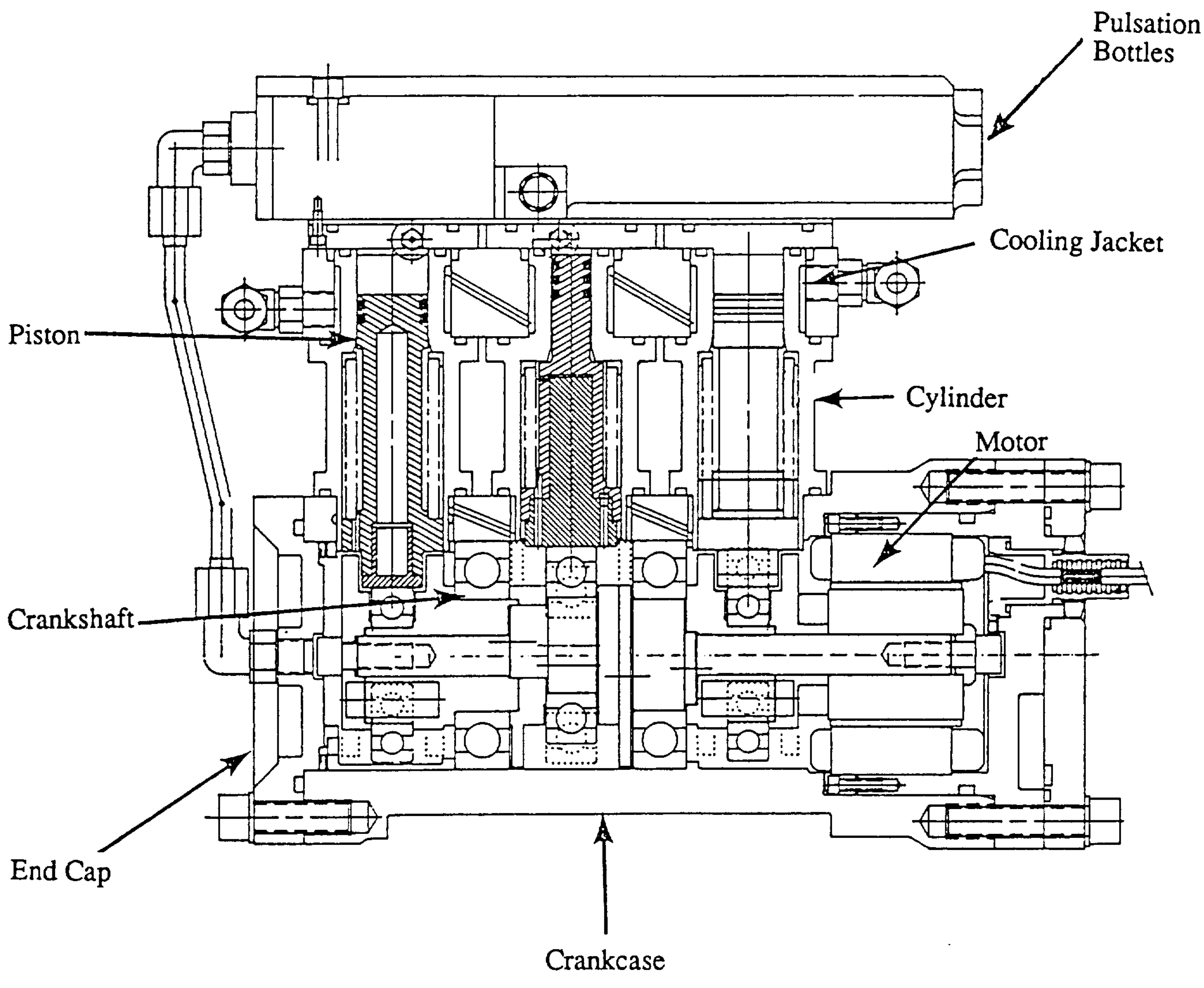


FIGURE 1. PROTOTYPE COMPRESSOR LAYOUT

The concept design for the remaining compressor types are based on this Type II design. The next section presents the design data and predicted performance.

6.0 EXAMPLE PREDICTED PERFORMANCE

Table II shows the basic conceptual design and performance information for the compressors for all seven applications which were outlined in Table I. The top section of Table II shows design data, including number of stages, bore, stroke, and cylinder clearance factor (CF). The full range of compression required by these seven applications is covered by three distinct cylinder sizes, or:

- 1 inch bore x 0.8 inch stroke
- 0.875 inch bore x 0.48 inch stroke
- 0.500 inch bore x 0.25 inch stroke

The compressor for the Type II application, which was carried to the prototype hardware stage as part of this project, consists of two stages. The first stage employs two identical cylinders with a bore of 0.875 inches and a stroke of 0.48 inches. The second stage has a single cylinder with a bore and stroke of 0.500 and 0.250 inches, respectively.

All other applications, with the exception of Type III, employ two stages with single cylinders per stage, each stage cylinder differing in bore and stroke. The Type I application employs identical cylinders for first and second stages (0.500 inch bore and 0.250 inch stroke) with proper loading accomplished by the selection of interstage pressure.

The second two sections of Table II show predicted performance data for each application. The suction, interstage, and discharge pressures are denoted by Ps, Pis, and Pd. Effective suction temperature at the cylinder intake valve (not the gas temperature at the compressor unit) is denoted by Ts.

Predicted flow rate is shown in the bottom portion of Table II for the conceptual design rotational speeds indicated. In general, each flow rate equals or slightly exceeds the values shown in Table I.

TABLE II. COMPRESSOR CONFIGURATION AND PREDICTED PERFORMANCE

Application	I	II	III	IV	V	VI	VII
Number of Stages	*2*	*2**	*1*	*2*	*2*	*2*	*2*
S1 - Bore mm (inches)	12.7 (.500)	22.22 (0.875)	22.22 (0.875)	22.22 (0.875)	22.22 (0.875)	25.4 (1.000)	22.22 (.875)
S1 - Stroke " "	6.35 (.250)	12.2 (0.480)	12.2 (0.480)	12.2 (0.480)	12.2 (0.480)	20.3 (0.800)	12.2 (.480)
S2 - Bore " "	12.7 (.500)	12.7 (0.500)	–	12.7 (0.500)	12.7 (0.500)	22.22 (0.875)	12.7 (.500)
S2 - Stroke " "	6.35 (.250)	6.35 (0.250)	–	6.35 (0.250)	6.35 (0.250)	12.2 (0.480)	6.35 (.250)
S1 - CF (%)	10	12	6	6	6	15	10
S2 - CF (%)	10	12	–	9	10	30	10
Ps - MPa (psia)	0.12 (18)	0.17 (25)	3.8 (550)	0.69 (100)	0.69 (100)	2.8 (400)	1.7 (250)
Pis - MPa (psia)	.59 (85)	1.64 (237)	–	4.1 (600)	4.1 (600)	10.4 (1500)	8.3 (1200)
Pd - MPa (psia)	8.27 (1200)	5.6 (815)	42.1 (6100)	20.7 (3000)	20.7 (3000)	41.4 (6000)	31.0 (4500)
Ts - °C (°F)	21 (70)	21 (70)	4.4 (40)	49 (120)	10 (50)	10 (50)	4.4 (40)
Speed - rpm	500	650	500	1250	1000	550	450
Flow Rate Kg/hr (lbm/hr)	.015 (.034)	.17 (0.37)**	8.4 (18.5)	.15 (0.33)	1.3 (2.8)	7.5 (16.6)	.23 (.51)

* The Type II compressor employs two first stage cylinders and one second stage cylinder with the dimensions shown. All other compressors employ one cylinder per stage.

** Based on observed first stage conditions.

7.0 SUMMARY AND CONCLUSIONS

This project has demonstrated the use of relatively slow speed, high ratio reciprocating compressor cylinders to satisfy the requirements of the Type II application. The other six applications may be accomplished by the conceptual designs discussed herein.

The predicted performance, total system weight, and volume requirements for the application types appear to fall well within the required specifications. The prototype compressor and drive design chosen should have long life and minimizes the risk of catastrophic failures.

APPENDIX B

PROTOTYPE DESIGN FINAL REPORT FOR ON-ORBIT COMPRESSOR TECHNOLOGY PROGRAM

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

1.0 INTRODUCTION

The objective of this project is the exploration of compressor technology applicable for use by the Space Station Fluid Management System (FMS), Space Station Propulsion System, and related on-orbit fluid transfer systems. The approach is to perform compressor research on a breadboard test article, to utilize the results to develop a conceptual design to handle seven different applications, and to develop a prototype for one of these specific applications. The prototype development will be based on the general conceptual design and consist of a detailed prototype design, fabrication of the prototype, and testing of the prototype. The primary emphasis is to develop basic compressor technology (designs, materials, and manufacturing techniques) in a time frame consistent with the support of the Space Station fluid systems development. Design considerations include: (1) maximization of service life; (2) commonalty; i.e., interchangeability of common hardware assemblies; (3) ease of maintenance; (4) lightweight; (5) small size; and (6) low power.

The purpose of this Prototype Design Final Report is to document the design of the prototype mixed gas compressor (Type II) for Space Station. The conceptual design is a two stage reciprocating piston type compressor with pressure actuated check valves.

2.0 DESIGN REQUIREMENTS

The prototype compressor developed under this project is intended for compressing an oxidizing gas mixture for storage/disposal on Space Station Freedom. The waste gas comes from the experiments on the Space Station and is discharged into a storage vessel initially at some minimum pressure until the vessel is pumped up to the upper design pressure. A complete list of design requirements are contained in the End Item Specification (EIS). Some of the general design specifications for compressor performance, operating environment, and working fluid are given below.

Compressor Performance Requirements	
Nominal Fluid Flow Rate	0.11 Kg/hr (0.25 LBM/hr)
Maximum Fluid Flow Rate	0.50 Kg/hr (1.10 LBM/hr)
Inlet Pressure	0.07 - 0.20 MPa (10 - 30 psia)
Discharge Pressure	0.69 - 6.90 MPa (100 - 1000 psia)
Maximum Discharge Pressure	8.28 MPa (1200 psia)
Inlet Fluid Temperature	15.5 - 32.2°C (60 - 90°F)
Duty Cycle	Continuous operation
Operating Life	10,000 hr

Operating Environment	
Weight Limitation	36.3 Kg (80 LBM)
Size Limitation	1.5 cubic feet
Power Limitation	1.0 KW Peak
Fluid Mixture	
Nitrogen	60.0%
Argon	19.3%
Oxygen	3.0%
Air	11.3%
Carbon Dioxide	1.9%
Krypton	1.8%
Xenon	0.6%
Helium	0.2%
Trace Reductants	<0.1%
Trace Contaminants	<1.8%
Inlet Dew Point	-30°F

3.0 INTERFACE DEFINITIONS

3.1 Fluid Connections

The inlet and discharge compressor working fluid lines are both 3/8 inch (EIS 4.2.7 Line Sizes) and the compressor will be welded into the fluid lines (EIS 3.1.1.1 Fluid). For the prototype compressor, the fluid fittings are O-ring sealed SAE straight thread fittings (7/16 - 20 thread).

3.2 Cooling Fluid Connections

No specific requirements for line size, temperature or flow rate are given in the EIS (see EIS 3.1.2.5). The prototype compressor is designed to use a 1/4-inch line connection of 10°C (50°F) cooling fluid. The heat rejection rate to the cooling fluid depends on the compressor operating conditions, but is estimated to be about 200 watts.

3.3 Electrical Interface

The only electrical connections required for the prototype compressor are for the drive motor and controller. This hardware is not being flight tested or qualified, but is being used to demonstrate that the commercially available brushless motor is capable of properly powering the compressor. The electrical interface requirements in the EIS are therefore not being implemented for the prototype. Details on the motor and controller are given in Section 7.3.

3.4 Mechanical Interface

The compressor will be bolted (EIS 3.1.1.2) to the mounting structure with 12 1/4-20 UNC bolts. The blind tapped mounting holes are located on both sides of the compressor case and the bottom face (see detailed drawing of case in Appendix A) for flexibility of installation.

4.0 DESIGN ANALYSIS

4.1 Conceptual Design

The compressor conceptual design is based on detailed performance modeling and breadboard testing (outlined in Section 5.1) to verify the model. The comprehensive digital time domain model includes cycle thermodynamics, real gas properties, in cylinder heat transfer, valve dynamics, ring leaks, piston friction, attached piping acoustic effects, and piston inertial effects. Besides predicting the compressor performance, the simulation model also provides prediction of several mechanical components such as: actuator bearing B-10 life, seal ring wear, actuator interface contact stress/life, valve stress/life, return spring stress/life, and piston loading.

The basic design, shown in Figure 1, is a 3-cylinder, two stage reciprocating piston type compressor with pressure actuated check valves. The pistons are follower actuated by eccentrically mounted anti-friction bearings. The piston is held in contact with the actuator with a preloaded spring. The following list presents the prototype compressor design parameters:

	First Stage		Second Stage
Number of Cylinders	2		1
Cylinder Bore (inches)	0.875		0.500
Compressor Nominal Speed (RPM)	650-1000		650-1000
Piston Displacement (cu. inches/cylinder)	0.288		0.049
Stroke (inches)	0.48		0.25
Clearance Volume (%)	6		10
Number of Suction Valves	2		1
Diameter of Suction Ports (inches)	0.125		0.094
Number of Discharge Ports	1		1
Diameter of Discharge Ports (inches)	0.125		0.094
Piston Guide Bore (inches)	1.250		1.250
Return Spring Preload (LB)	15		15
Return Spring Rate (LB/inch)	60		60
Actuator Bearing Size	206		106
Crank Main Bearing Size		207	
Motor Peak Rated Torque (oz.-in.)		400	
Motor Power at Rated Peak Torque (watts)		510	
Maximum Continuous Output Power (watts)		560	

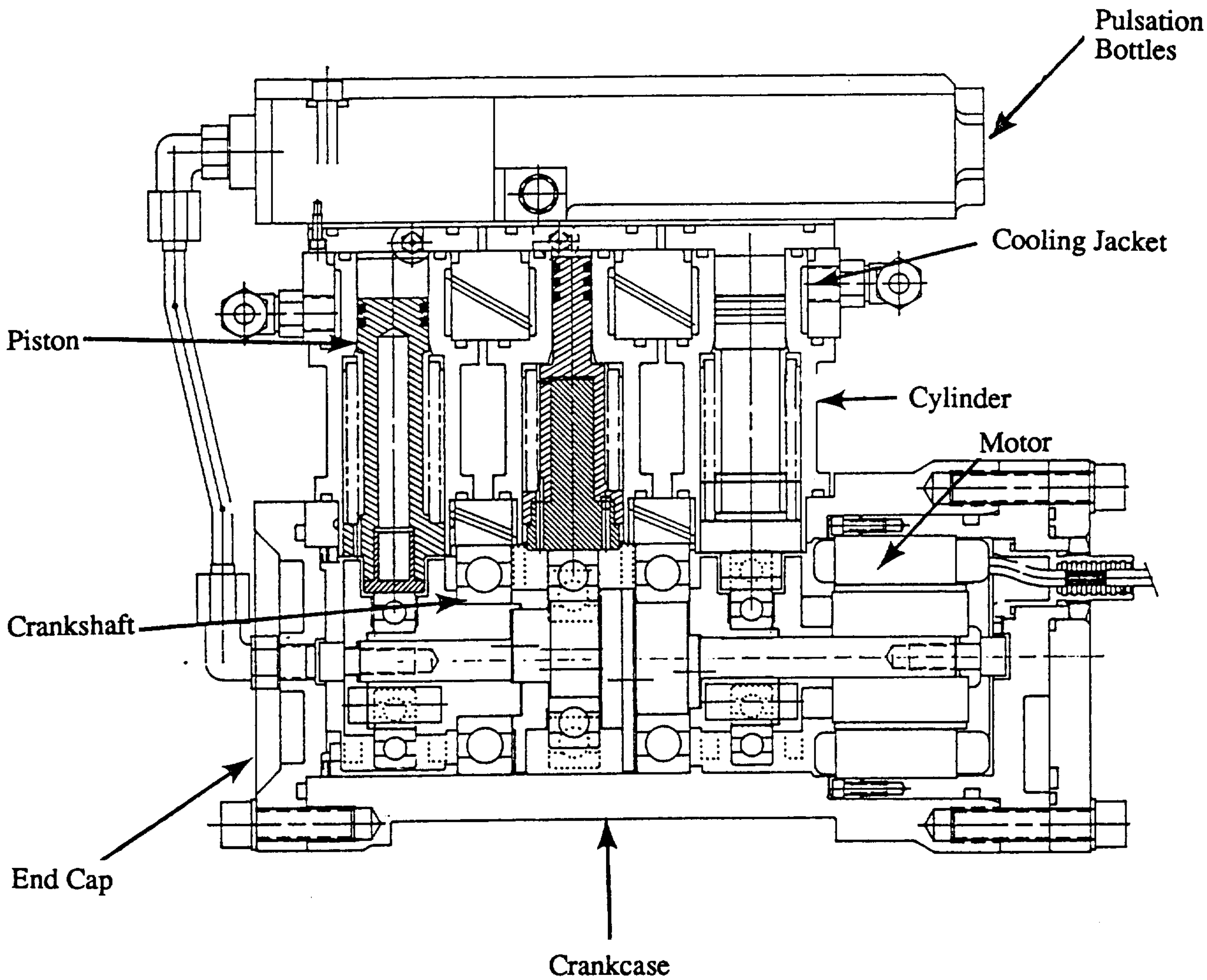


FIGURE 1. PROTOTYPE COMPRESSOR LAYOUT

The above outlined design was based on a number of competing design requirements and represents a reasonable trade-off between performance, reliability and life requirements. The design is simple with few moving parts, and based on component wear and life predictions, is free of sudden catastrophic failure modes. Both compressor stages are driven from a single drive motor and crank assembly resulting in fewer mechanical components and lighter weight compared with separate stages. The integration of both stages also simplifies installation since manifolding between the two stages and the pulsation bottles is incorporated into a single head assembly.

The three cylinder design is balanced to eliminate primary and residual secondary shaking forces. While the three cylinder design is somewhat more complicated than other possible designs, the ability to limit shaking forces is very important. The option of an unbalanced compressor with compensating hardware (active or passive devices) was investigated and determined to be unacceptable for a variable speed compressor.

4.2 Mechanical Design

4.2.1 Overall Design Layout

Each of the major mechanical subassemblies will be described in this section. Along with the descriptions, some of the rationale behind the design and some of the major design trade-offs will be discussed. Material properties of each of the components is discussed in Section 4.3. Each of the compressor components is labeled in the layout drawing shown in Figure 1. Appendix A contains the detailed fabrication drawings for each of the components discussed in this section.

4.2.2 Crankcase

The crankcase subassembly is shown in Figure 2 and consists primarily of the crankcase housing and the two end caps. Several other components are contained in the crankcase subassembly in order to retain the motor and crank main bearings and provide for the motor wire feedthrough. The piston cylinders are bolted to the top face of the crankcase and recesses cut into the case allow the cylinder to protrude below the case surface. This reduces the required piston length and weight by closely coupling the base of the piston to the actuator. Three sets of blind taped holes are provided on the case to allow the compressor to be mounted in any orientation.

The primary design constraint affecting the crankcase design is the case must withstand a burst pressure of 250% (2500 psi) of the maximum discharge pressure. This requirement results in a thick case wall, thick end caps and requires large bolts for attaching the end caps. Sufficient case stiffness could be attained with substantially less weight if the case design pressure was reduced since the case pressure is vented to the compressor suction port.

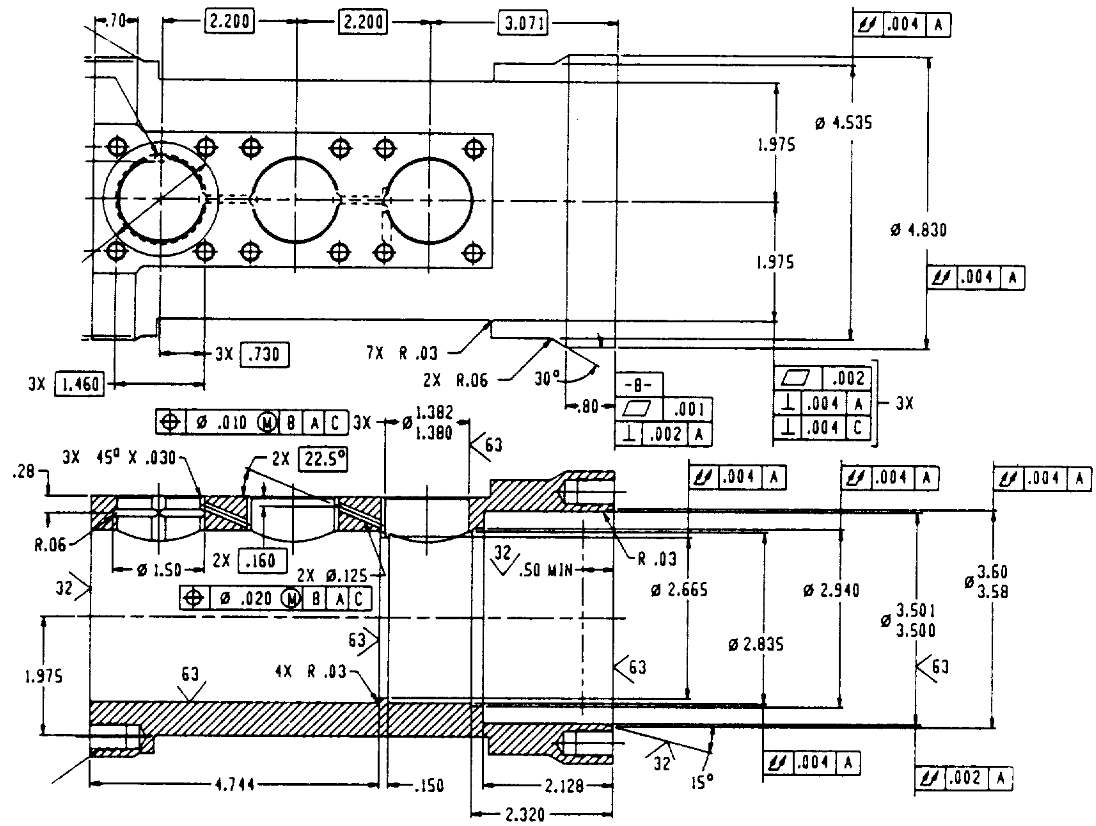


FIGURE 2. CRANKCASE DESIGN

4.2.3 Crankshaft

The crank subassembly is shown in Figure 3 and consists of the piston actuator bearings, crank main bearings, the crank itself and the mount for the motor rotor. The compound crankshaft consists of 6 mating parts that slide onto the crankshaft and are held in place by end bolts. The bearings are lightly press-fit onto the crankshaft and are securely held in place by shoulders on either side of the inner bearing race. When the crank assembly is placed into the crankcase, the main bearings are held in place by one fixed shoulder and a retainer ring acting on the bearing outer race. The spring preload on the actuator bearing keeps the main bearings firmly seated on the bottom of the case.

The bearings selected for the prototype compressor are standard grease filled bearings with 52100 races and balls, phenolic cages, and elastomeric seals. The following table lists the bearings and their size. The bearing size requirements were dictated by the required piston stroke, which results in more than adequate load ratings. The predicted minimum life rating for the bearings at the design conditions is 250,000 hours. The selection of ball type bearings was based on their ability to tolerate slight misalignments and handle both radial and axial loads.

	Part #	ID x OD x W (in.)
Main Bearings	207	1.378 x 2.835 x 0.67
First Stage Actuator	106	1.181 x 2.165 x 0.51
Second Stage Actuator	206	1.181 x 2.441 x 0.63

The primary design requirements that dictated the crank design were: minimization of vibration, reliability and long life, ease of assembly and disassembly, timing between stages must be maintained (eliminate the possibility of slipping between crank components) and to provide as compact and lightweight a design as possible. To accomplish these design requirements, a cam follower design was selected over other candidates because of simplicity, reliability, and elimination of shaking forces.

Considerable effort went into counterbalancing the relatively large, eccentrically mounted actuator bearings to eliminate vibration. Each individual component of the compound crank was carefully designed to balance the crank with the least possible weight addition. To eliminate the possibility of relative movement between crank components, the components were made to precisely mate in only one orientation.

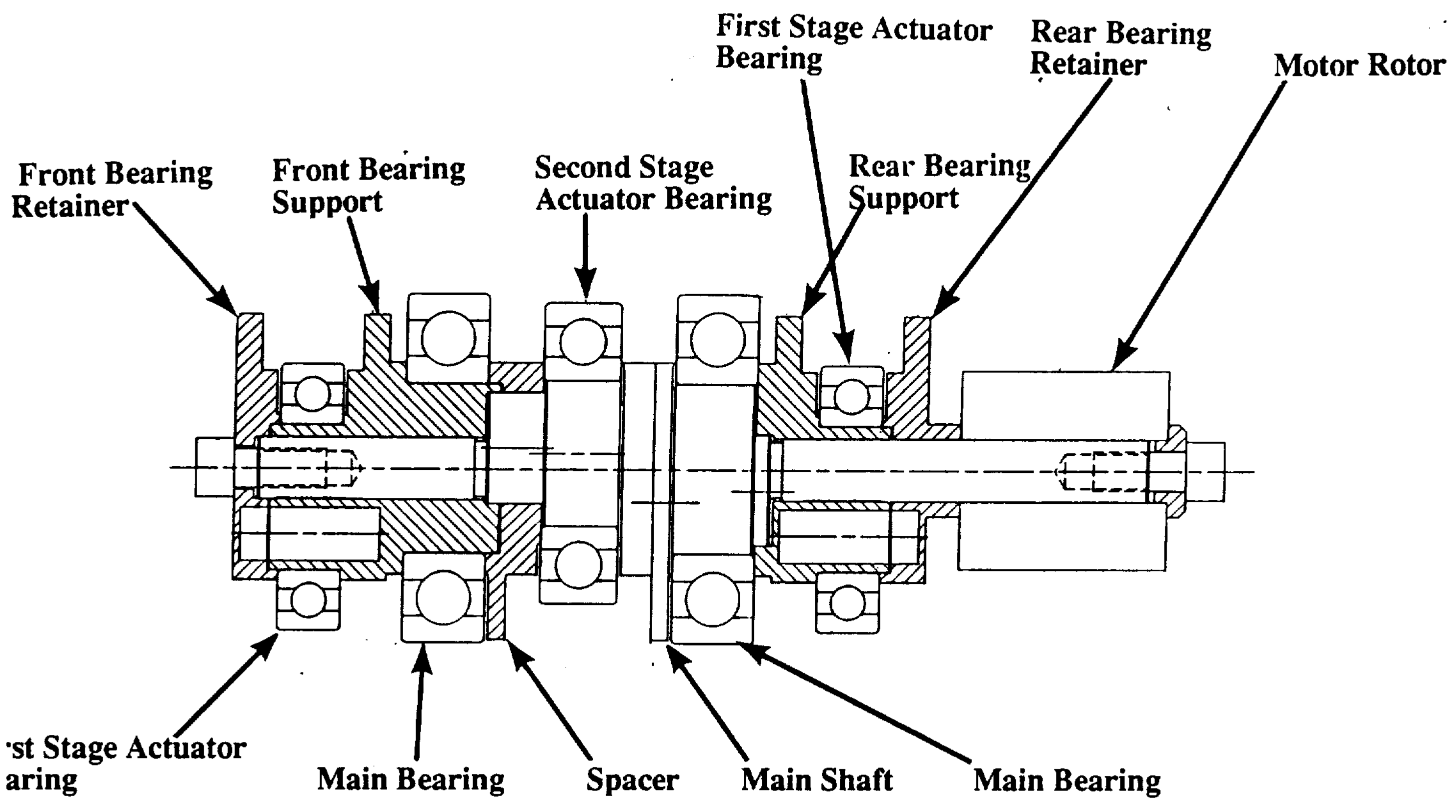


FIGURE 3. CRANKSHAFT DESIGN

4.2.4 Pistons

The compressor pistons are composed of the piston with integral upper and lower guide rings, seal rings, and the cam follower. The first stage piston is shown in Figure 4 and is sized for a 0.875 inch diameter bore while the second stage is 0.500 inch. Both pistons have a 1.125 inch diameter lower guide ring that provides lateral piston support in addition to a shoulder for the return spring to act on. Inserted in the base of each piston is a cam follower that provides a long life wear surface in contact with the outer bearing race used as an actuator.

In order to balance the shaking forces between the first and second stage pistons (which operate 180 degrees out of phase), the second stage piston weighs approximately four times one of the first stage piston assemblies. In order to accomplish this, the first stage piston assemblies are as light as possible while a weighted cam follower is provided for the second stage. The pistons are made of Torlon 4301 material to provide a strong, lightweight piston with side walls that provide integral guide rings. Tests have shown that separate guide rings made of Rulon F (Dixon) or Turcite (Shamban) may be superior to the Torlon (Amoco) for wear resistance and long life.

After reviewing and testing several seal designs, it became apparent the requirements for long life, no lubrication, mixed gas environment, and high pressure and velocity conditions precluded the use of commercially available seals. The seal design needed to have a thick wear surface for long useful life, a spring to keep the seal in contact with the cylinder, a low spring rate (to minimize the seal pressure on the cylinder) with large deflection, an overall low seal dead volume, low leakage, ease of installation, and must be made of low wear stable material. The seal design selected consists of a simple Rulon F ring that is O-ring energized. Detailed drawings of the seals and springs are given in Appendix A and Section 5.2 describes the seal testing.

The cam follower for the first stage pistons is made of 440-C stainless steel hardened to 55 Rc. The second stage cam follower is made of tungsten which provides a hard contact surface in addition to the high density required for the counterweight. The cam followers and pistons have vent holes through them to allow the spring cavity to vent to the crankcase.

4.2.5 Cylinders

The compressor's three cylinders are individually machined cylinders that are bolted to the top of the crankcase. The first stage cylinders are shown in Figure 5 and have an upper bore of 0.875 inches and a lower bore of 1.125 inches to accommodate the return spring. The second stage cylinder has a 0.500 inch upper bore and a 1.125 inch lower bore. The cylinders are made of 6061-T6 aluminum with a Tiodize Hardtuf X20 surface treatment for low wear and compatibility with the

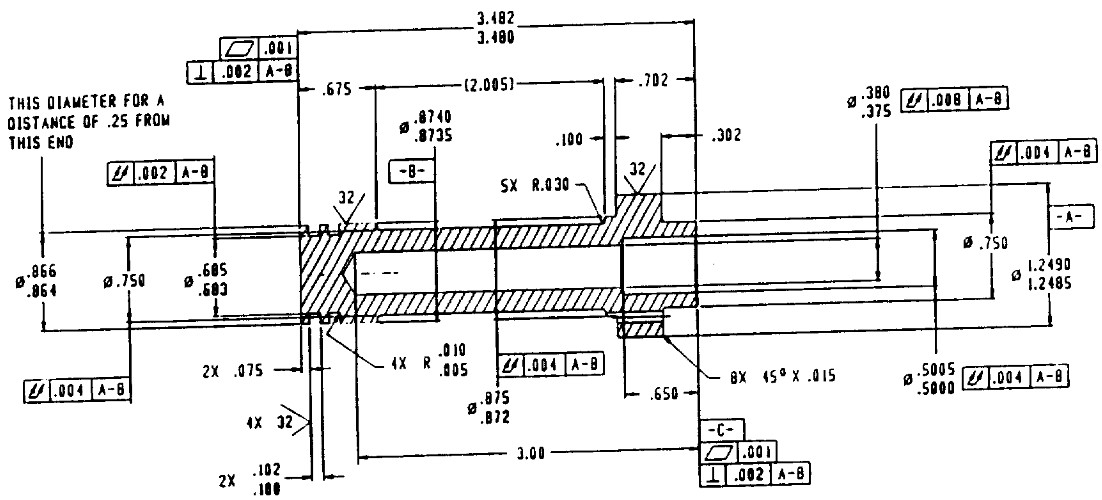


FIGURE 4. PISTON DESIGN

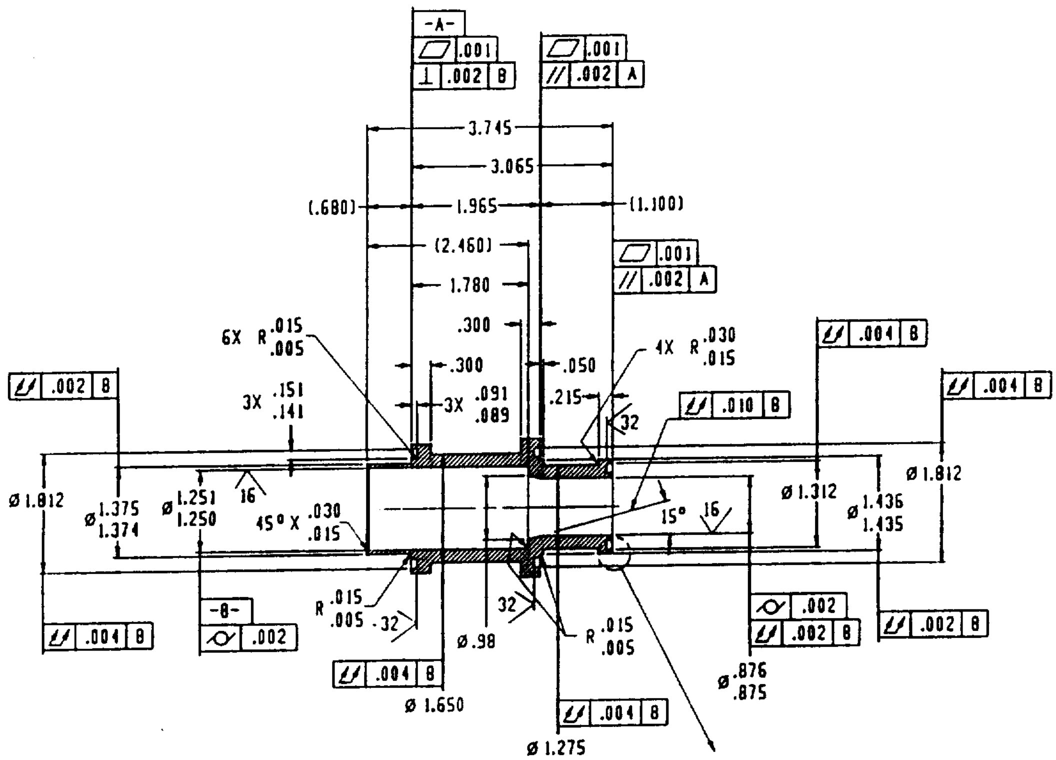


FIGURE 5. CYLINDER DESIGN

gas mixture. This treatment produces a low friction, hard, corrosion resistant, and wear resistant surface. The process is similar to aluminum anodizing, but a fluorocarbon polymer is infused into the surface.

The decision to use individual cylinders instead of a "block" provided a lighter design and simplified manufacturing and assembly. Individual cylinders also allowed a simple cooling jacket to provide uniform cooling fluid distribution over the cylinder. This design eliminated the need for sleeves in the compressor and the associated sealing issues. The use of aluminum with a thin surface treatment allowed good heat transfer through the cylinder wall to the cooling fluid.

4.2.6 Valves

The compressor valve assemblies consist of an aluminum valve plate (as shown in Figure 6), the stainless steel check valves (as shown in Figure 7), seals, and aluminum valve retainers. Once assembled, the valve subassembly can easily be installed and removed from the compressor. Face seals are used between the cylinder and valve plate and between the valve plate and the head. The suction and discharge passages through the valve plate are also separated by face seals.

To enhance heat transfer within the cylinder, the first stage contains two suction valves oriented to produce swirl in the cylinder. The smaller second stage cylinder has room for only one suction and one discharge valve, but the surface area to volume ratio in the second stage allows good gas contact with the cooled cylinder walls. As with the second stage, the first stage has one discharge valve.

To produce a long life, low pressure drop, reliable pressure actuated check valve design, the valves were made relatively long to reduce stress levels and pressure drop. With a long valve and the low clearance volume, the valve retainers had to be recessed into the valve plate so they would clear the piston. A recess could not be cut into the piston for clearance since the piston is free to rotate. A "sheet valve" design was considered, but due to the close tolerance on valve alignment, low piston to head clearance (thermally induced bowing), and the need for large seals above and below each sheet valve, individual valves were selected. The selection of individually mounted valves will provide the most reliable valve design and allow easy compressor assembly.

4.2.7 Cylinder Head

The cylinder head (as shown in Figure 8) contains the pulsation bottles, cooling passages, and gas manifolding to all three cylinders. The pulsation bottles are sized 20 times the swept piston volume to filter pulsation from passing into the attached piping and reduce pressure fluctuations that adversely affect compressor performance. By integrating the bottle into the head, the piping

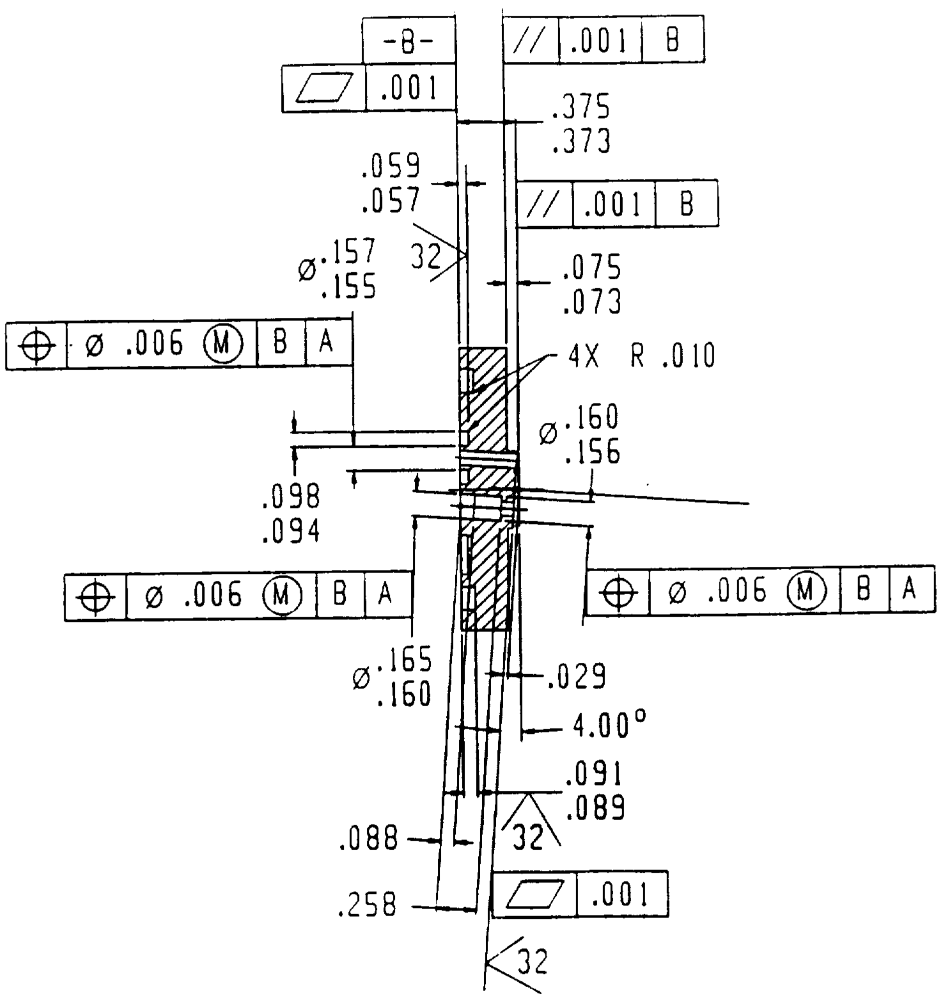


FIGURE 6. VALVE PLATE DESIGN

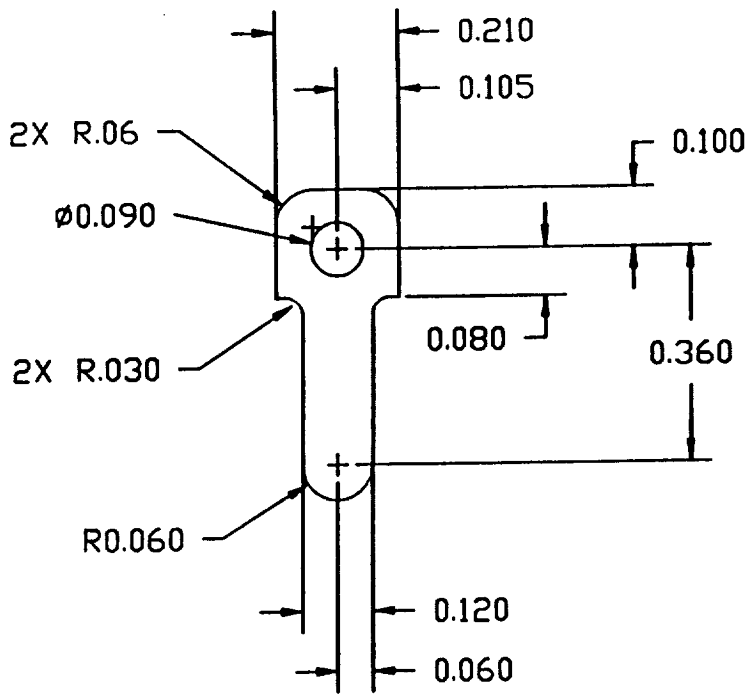


FIGURE 7. CHECK VALVE DESIGN

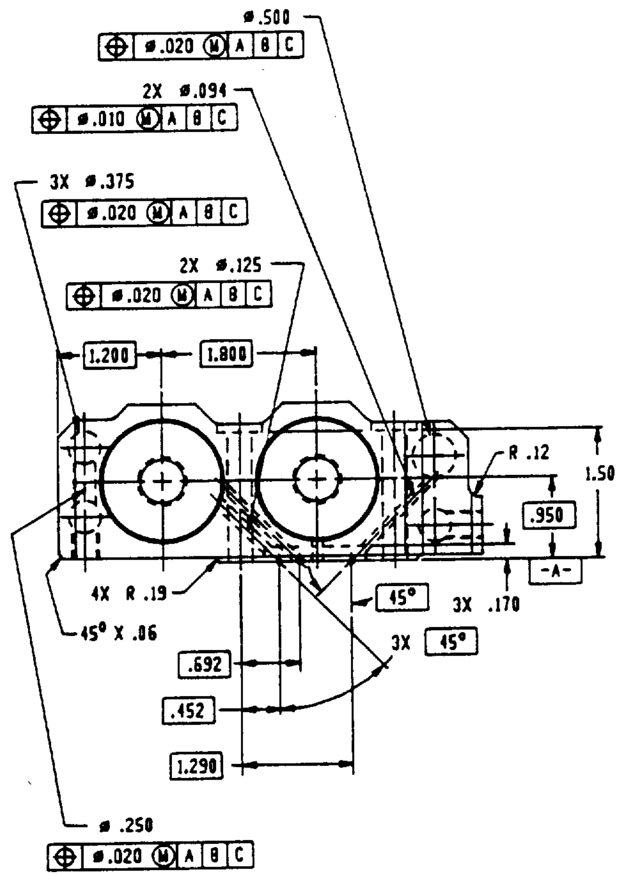


FIGURE 8. CYLINDER HEAD DESIGN

lengths were shortened which reduces pressure drop and more closely couples the bottles to the cylinders. Cooling channels in the head provide cooling for the gas as it passes through the manifold and bottles.

The head is made of 6061-T6 clear anodized aluminum that is a lightweight, strong material with high thermal conductivity. The compressor inlet and discharge lines connect to one end of the pulsation bottles, and the other end is closed with an aluminum plug retained with a snap ring. The cooling fluid channels are located next to the discharge and inner-stage bottles to remove heat generated in the compressor. The cooling fluid used in the prototype compressor is ethelyene glycol, but other fluids that are compatible with anodized aluminum can be used.

4.2.8 Cooling Jacket

The cooling jacket is shown in Figure 9 and fits over the top portion of the cylinders and provides a passage for the cooling liquid. The fluid flows from a first stage cylinder to the second stage cylinder and then to the other first stage cylinder. Because the heat rejection rate is relatively low (less than 200 watts), the temperature gradient in the fluid will be negligible. The cooling jacket is made of 6061-T6 clear anodized aluminum for high thermal conductivity, good strength, lightweight and ease of machining.

4.3 Material Specifications

4.3.1 Compressor Materials

Table I presents the Material Identification and Usage List and is a summary of the materials used in the prototype compressor. The materials in the drive motor and controller are not included for reasons discussed in Section 7.3.

4.3.2 Trace Contaminates

4.3.2.1 Approach

The trace contaminates trade study consists of a series of tables which summarize the engineering review of the material compatibility for the Space Station waste gas compressor. This review was based on the assumption that the following environmental conditions applied:

<u>Temperature Range</u> - 60° - 120°F (Temperatures up to 250°F were considered in the review)

<u>Pressure Range</u> - 10 - 30 psia (Suction, 1st stage)
100 - 1200 psia (Discharge, 2nd stage)

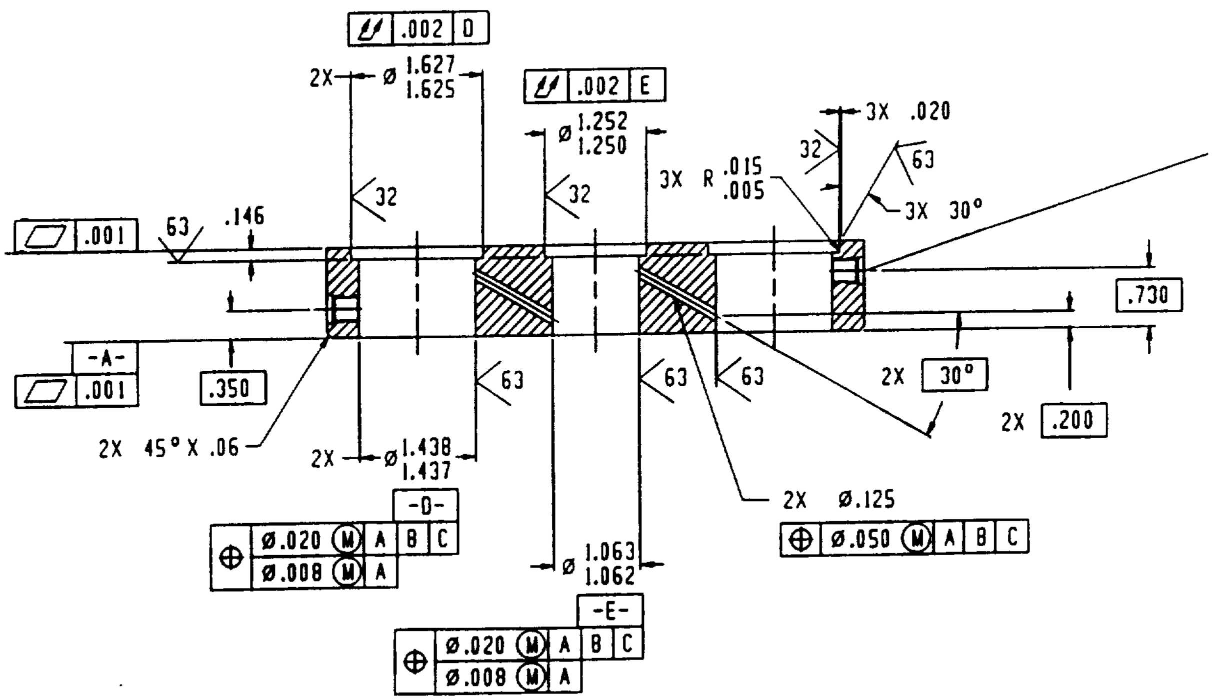


FIGURE 9. COOLING JACKET DESIGN

TABLE I. MATERIAL IDENTIFICATION AND USAGE LIST

PART DESCRIPTION	MATERIAL DESCRIPTION	MATERIAL MANUFACTURER	SPECIFICATIONS
Flange, Front	Aluminum, 6061-T6		Anodize per MIL-A-8625C, Type 2, Class I
Spacer, Bearing	Aluminum, 6061-T6		Anodize per MIL-A-8625C, Type 2, Class I
Retainer, Motor	Aluminum, 6061-T6		Anodize per MIL-A-8625C, Type 2, Class I
Washer, Motor	SST, Type 304		Passivate per MIL-STD-171C, Finish 5.4.1
Retainer, Front Bearing	SST, Type 304		Passivate per MIL-STD-171C, Finish 5.4.1
Retainer, Rear Bearing	SST, Type 304		Passivate per MIL-STD-171C, Finish 5.4.1
Spacer, Crankshaft	SST, Type 304		Passivate per MIL-STD-171C, Finish 5.4.1
Bearing Support, Front	SST, Type 304		Passivate per MIL-STD-171C, Finish 5.4.1
Bearing Support, Rear	SST, Type 304		Passivate per MIL-STD-171C, Finish 5.4.1
Main Shaft	SST, Type 304		Passivate per MIL-STD-171C, Finish 5.4.1
Feedthrough, Electrical	Aluminum, 6061-T6		Anodize per MIL-A-8625C, Type 2, Class I
Nut, Feedthrough	Aluminum, 6061-T6		Anodize per MIL-A-8625C, Type 2, Class I

TABLE I (Continued). MATERIAL IDENTIFICATION AND USAGE LIST

PART DESCRIPTION	MATERIAL DESCRIPTION	MATERIAL MANUFACTURER	SPECIFICATIONS
Cylinder, First Stage	Aluminum, 6061-T6	Treatment: Tiodize Co., Inc. Huntington Beach, CA	Treat per MIL-A-6357A (AR) Approved Source. Tiodize "Hardtuf X20" Process
Cylinder, Second Stage	Aluminum, 6061-T6	Treatment: Tiodize Co., Inc. Huntington Beach, CA	Treat per MIL-A-6357A (AR) Approved Source. Tiodize "Hardtuf X20" Process
Piston, First Stage	Torlon 4301	Amoco Chemical Corp.	
Piston, Second Stage	Torlon 4301	Amoco Chemical Corp.	
Follower, First Stage Piston	SST, Type 440-C		Hardening Temp 1850-1950 F. Air Cool. Draw at 800 F for 1 hr. RC 55-58
Follower, Second Stage Piston	Tungsten Alloy, 97% W, 2.1% Ni, 0.9% Fe	Mi-Tech Metals, Indianapolis, IN	
Seat, Piston Spring	Aluminum, 6061-T6		Anodize per MIL-STD-8625C, Type 2, Class I
Pin, Piston Locating	SST, Type 304		Passivate per MIL-STD-171C, Finish 5.4.1
Return Spring, Piston	SST, Type 302		Shot Peen
Check Valve	SST, Type 302		Flat Smooth Stock, Full Hard, QQ-S-766
Seal	Rulon F	Dixon Industries Corp.	
Potting Matl, Electrical Feedthrough	Epoxy "C-7" with "W" Activator	Armstrong Products	

TABLE I (Continued). MATERIAL IDENTIFICATION AND USAGE LIST

PART DESCRIPTION	MATERIAL DESCRIPTION	MATERIAL MANUFACTURER	SPECIFICATIONS
Retainer, First Stage Intake Valve	Aluminum, 6061-T6		Anodize per MIL-STD-8625C, Type 2, Class 1
Retainer, First Stage, Outlet Valve	Aluminum, 6061-T6		Anodize per MIL-STD-8625C, Type 2, Class 1
Nut, First Stage Inlet Valve Retainer	Aluminum, 6061-T6		Anodize per MIL-STD-8625C, Type 2, Class 1
Retainer, Second Stage Outlet Valve	Aluminum, 6061-T6		Anodize per MIL-STD-8625C, Type 2, Class 1
Retainer, Second Stage Inlet Valve	Aluminum, 6061-T6		Anodize per MIL-STD-8625C, Type 2, Class 1
Valve Plate, First Stage	Aluminum, 6061-T6		Anodize per MIL-STD-8625C, Type 2, Class 1
Valve Plate, Second Stage	Aluminum, 6061-T6		Anodize per MIL-STD-8625C, Type 2, Class I
Valve Plate, First Stage	Aluminum, 6061-T6		Anodize per MIL-STD-8625C, Type 2, Class 1
Valve Plate, Second Stage	Aluminum, 6061-T6		Anodize per MIL-STD-8625C, Type 2, Class 1
Manifold, Cylinder Cooling Fluid	Aluminum, 6061-T6		Anodize per MIL-STD-8625C, Type 2, Class 1
Pulsation Chamber	Aluminum, 6061-T6		Anodize per MIL-STD-8625C, Type 2, Class 1
Cooling Plate, Motor	Aluminum, 6061-T6		Anodize per MIL-STD-8625C, Type 2, Class 1

Atmosphere - A mixture of the species listed in Group 1.2 (Oxidizing Gas Mixture) and in Group 2.0 (Space Station IWFS Potential Trace Contaminants) of Appendix 2: "Entrained Vapors and Trace Contaminants", of the EIS - Revision 1.32.

The materials considered during this compatibility study were those proposed for the primary compressor components which will be exposed to the gas path. The components considered and the materials of construction were as follows:

Compressor Cylinder

The prototype compressor unit is fabricated from a 6061-T6 aluminum cylinder, which has been given a Hardtuf™ X20 surface treatment (a registered trademark of Tiodize Co., Inc.). This treatment involves anodizing of the aluminum pieces followed by impregnation of the porous anodized layer with Teflon (PTFE). To account for the possibility that the Teflon may eventually be pulled out of the anodized layer over time, due to wear, the compatibility of both PTFE and aluminum with the waste gas flow stream was evaluated.

Compressor Rings

- Polyamide-Imide (Torlon®, a product of Amoco Chemicals Corp.)
- PTFE (Rulon®)

Compressor Piston

- Hardtuf™ Treated Aluminum
- Polyamide-Imide (Torlon®)

Valves

- Stainless Steel Type 302

4.3.2.2 Method of Evaluation

The compatibility of the various materials with the waste gas flow stream was evaluated by utilizing data from the open literature, vendor supplied information, and personal experience of the Institute Staff. The potential effects of the individual waste gas flow stream species were assessed, and each material/species combination was rated as being compatible, incompatible, or as having insufficient data available. The synergistic effects of a combination of two or more species, or the compatibility of the materials with new species which may result from a reaction between two or more components within the flow stream, were not considered. To evaluate the potential detrimental effects of these higher order reactions would require a much more extensive effort, and most likely a laboratory testing program.

4.3.2.3 Results

The results of the material compatibility study are presented in Tables II-V.

4.3.2.4 Discussion

The literature available for assessment of the compatibility between a material and potential waste gas flow stream species is generally data from testing in relatively high concentrations of the subject species as compared to the concentrations expected in the Space Station Waste Gas System. In excess of 95% of the flow stream is expected to be inert gases. The dilution effect by this high mass percentage of inert species should reduce the corrosive nature of most of the species ranked as incompatible, and in fact allow them to be handled with no resulting damage to the compressor.

A number of the species ranked as incompatible are relatively benign unless there is water present. The most likely scenario for corrosion damage to the compressor involves the formation of free condensed water. Most of the incompatible vapor species will tend to be absorbed by, and concentrated in any liquid water phase which forms, allowing active corrosion to occur. The most severe corrosion would be expected to occur during any down time in the compressor's duty cycle, when the residence time of any condensed species could be relatively long. Since many of the waste gas species are obviously vapors from aqueous solutions, it would most likely be unrealistic to dehydrate the flow stream. It may, however, be advantageous to purge the compressor by running dried inert gases through it prior to any down portions of the cycle.

The fine dust particles listed were evaluated based on their potential for causing corrosion damage. If particles of the dust were able to penetrate the Hardtuf™ layer, and become imbedded, forming a metallic contact with the cylinder, it is probable that a number of the species could cause pitting of the aluminum. All of the metallic species except beryllium and cadmium would most likely be significantly cathodic to the aluminum, and could cause pits due to galvanic effects. The metal/halide compounds, if imbedded and in the presence of water, could act as a source of free halide ions, which are known to cause localized attack of both aluminum and most stainless steel alloys.

4.3.2.5 Summary

In general, the majority of the species which are possible in the waste gas flow stream are not aggressive to the materials of construction of the compressor. The most corrosive species are the halide gases, ammonia, and the vapors from the strong acids and bases. Even these species are

TABLE II. COMPATIBILITY OF ALUMINUM ALLOYS WITH THE WASTE GAS FLOW STREAM

	COMPATIBLE	INCOMPATIBLE	INSUFFICIENT DATA
OXIDIZING GAS MIXTURE	Nitrogen Argon Oxygen Carbon Dioxide Krypton Hydrogen Xenon Helium		
GASES	Acetylene Carbon Monoxide Ethylene Freon 22 Propane	Ammonia(1) Chlorine Fluorine	Nitrogen Dioxide Silicon Hydride
ETCHANT SOLUTIONS	Hydrofluoric Acid(2) Hydrogen Peroxide Methanol Potassium Ferricyanide	Acetic Acid Bromine Hydrochloric Acid Nitric Acid Perchloric Acid Potassium Hydroxide Silver Nitrate Sodium Hydroxide Sodium Hypochlorite	Cupric Nitrate Magnesium Iodide
SOLVENTS	Acetone Benzene Trichloroethylene		
ORGANIC OTHER	Acrolein Allyl Alcohol Butyl Lactate Chlorodifluoroethane Cyclohexanol Dichloromethane Diisobutyl Ketone Isopropyl Alcohol Methyl Ethyl Ketone N-Butyl Alcohol Toluene Trichloroethane Trichlorotrifluoroethane Trimethylbenzene Xylene		Furan Glutaraldehyde Indene Phenol Polyphenylene Sulfides Triglycene Sulfate
INORGANICS OTHER	Water(4)	Ammonia Mercury(3)	
FINE DUST	Beryllium Cadmium Germanium(2) Lithium Niobium(2) Silicon Tellurium(2) Tungsten(2)		Cadmium Sulfide Gallium Arsenide Magnesium Iodide(2) Mercurous Iodide(2) Sodium Aluminate Sodium Chlorate(2)

(1) Compatible if system remains free of condensed water
(2) May cause pitting
(3) Possible SCC
(4) Species going into solution in condensed water is the most likely mechanism for attack by many of the incompatible species, and could cause a number of species listed as compatible to become aggressive. Preventing the condensation of water would help prevent many of the potential incompatability problems.

TABLE III. COMPATIBILITY OF STAINLESS STEEL TYPE 302 WITH THE WASTE GAS FLOW STREAM

	COMPATIBLE	INCOMPATIBLE	INSUFFICIENT DATA
OXIDIZING GAS MIXTURE	Nitrogen Argon Oxygen Carbon Dioxide[1] Krypton Hydrogen Xenon Helium		
GASES	Acetylene Ammonia Carbon Monoxide Ethylene Freon 22 Propane	Chlorine Fluorine	Nitrogen Dioxide Silicon Hydride
ETCHANT SOLUTIONS	Hydrofluoric Acid[2] Hydrogen Peroxide Methanol Potassium Ferricyanide Potassium Hydroxide Silver Nitrate Sodium Hydroxide	Acetic Acid Bromine Hydrochloric Acid Magnesium Iodide Nitric Acid Perchloric Acid Sodium Hypochlorite	Cupric Nitrate
SOLVENTS	Acetone Benzene Trichloroethylene		
ORGANIC OTHER	Acrolein Allyl Alcohol Chlorodifluoroethane Cyclohexanol Dichloromethane Diisobutyl Ketone Isopropyl Alcohol Methyl Ethyl Ketone N-Butyl Alcohol Toluene Trichloroethane Trichlorotrifluoroethane Trimethylbenzene Xylene		Furan Glutaraldehyde Indene Phenol Polyphenylene Sulfides Triglycene Sulfate
INORGANICS OTHER	Ammonia Mercury Water[3]		
FINE DUST	Beryllium Cadmium Germanium Lithium Niobium Silicon Tellurium Tungsten		Cadmium Sulfide Gallium Arsenide Magnesium Iodide Mercurous Iodide Sodium Aluminate Sodium Chlorate

(1) Compatible if system remains free of condensed water
(2) May cause pitting
(3) Species going into solution in condensed water is the most likely mechanism for attack by many of the incompatible species, and could cause a number of species listed as compatible to become aggressive. Preventing the condensation of water would help prevent many of the potential incompatability problems.

TABLE IV. COMPATIBILITY OF POLYAMIDE-IMIDE (TORLON®) WITH THE WASTE GAS FLOW STREAM.

	COMPATIBLE	INCOMPATIBLE	INSUFFICIENT DATA
GASES	Acetylene Argon Carbon Dioxide Carbon Monoxide Chlorine Ethylene Fluorine Freon 22 Helium Hydrogen Krypton Nitrogen Dioxide Oxygen Propane Silicon Hydride Xenon	Ammonia	
ETCHANT SOLUTIONS	Acetic Acid Bromine Cupric Nitrate Magnesium Iodide Methanol Potassium Ferricyanide Silver Nitrate Sodium Hypochlorite	Hydrochloric Acid Hydrofluoric Acid Nitric Acid Perchloric Acid Potassium Hydroxide Sodium Hydroxide	Hydrogen Peroxide
SOLVENTS	Acetone Trichloroethylene	Benzene	
ORGANIC OTHER	Acrolein Butyl Lactate Chlorodifluoroethane Cyclohexanol Dichloromethane Diisobutyl Ketone Furan Glutaraldehyde Indene Isopropyl Alcohol Methyl Ethyl Ketone Phenol Polyphenylene Sulfides Toluene Trichloroethane Trichlorotrifluoroethane Triglycene Sulfate Xylene		Allyl Alcohol N-Butyl Alcohol Trimethylbenzene
INORGANICS OTHER	Mercury Water	Ammonia	
FINE DUST	All		

TABLE V. COMPATIBILITY OF PTFE (TEFLON) WITH THE WASTE GAS FLOW STREAM.

	COMPATIBLE	INCOMPATIBLE	INSUFFICIENT DATA
GASES	All, Except Fluorine	Fluorine	
ETCHANT SOLUTIONS	All		
SOLVENTS	All		
ORGANIC OTHER	All		
INORGANIC OTHER	All		
FIND DUST	All		

expected to cause minimal damage due to the dilution effects from the large volume of inert gases which are expected to comprise over 95% of the flow stream. Even the potential for material degradation due to the synergistic action of mixtures of the more hazardous species should be minimized by the dilution effects.

The corrosive potential of most of the aggressive species can be dramatically increased by the presence of condensed water. It will be important to minimize the potential for water formation. It is also recommended that the compressor be purged with dry inert gas, especially prior to any down periods in the duty cycle. This will effectively minimize the residence time of any liquid water phase in which aggressive waste gas species could dissolve.

5.0 RESULTS FROM SUBASSEMBLY TESTING

5.1 Breadboard Test Article

5.1.1 Performance Testing Objectives

The objective of the Breadboard Test Article (BTA) activity is to investigate the effect of active cooling of the compressor cylinder wall on overall compressor performance. Secondary parameters investigated are the effect of compressor valve location and pulsation control. The data obtained in these tests helped verify and update the cylinder heat transfer models used in the compressor simulation code, which is the primary compressor design tool.

5.1.2 Performance Test Apparatus

The BTA is shown in Figure 10 and is a modified prime mover with the cylinder and head layout illustrated in Figure 11. Two different valve configurations were tested to look at the effects of increased cylinder swirl on performance. The first valves tested have one inlet and one discharge port with no attempt in the valve design to increase mixing and heat transfer rates. The first set of valves tested are flat Reed valves with three inlet and three discharge valves. These valves are located near the wall to increase turbulence and heat transfer to the wall. The second set of valves tested also are Reed valves with three inlet and three discharge valves (in the same location as above), but the valves are positioned to impart a radial swirl to the incoming gas in an attempt to further increase heat transfer. Both valve types are shown in Figure 12.

Cylinder cooling is accomplished by flowing cooled compressed air around the outside of the cylinder jacket. The compressor is driven with a variable speed DC motor capable of a maximum speed of 4000 RPM. The motor is mounted in a fixture to allow torque measurements and a shaft encoder is incorporated onto the drive shaft to measure angular position of the compressor crank. Additional instrumentation is provided to measure suction and discharge pressure and temperature,

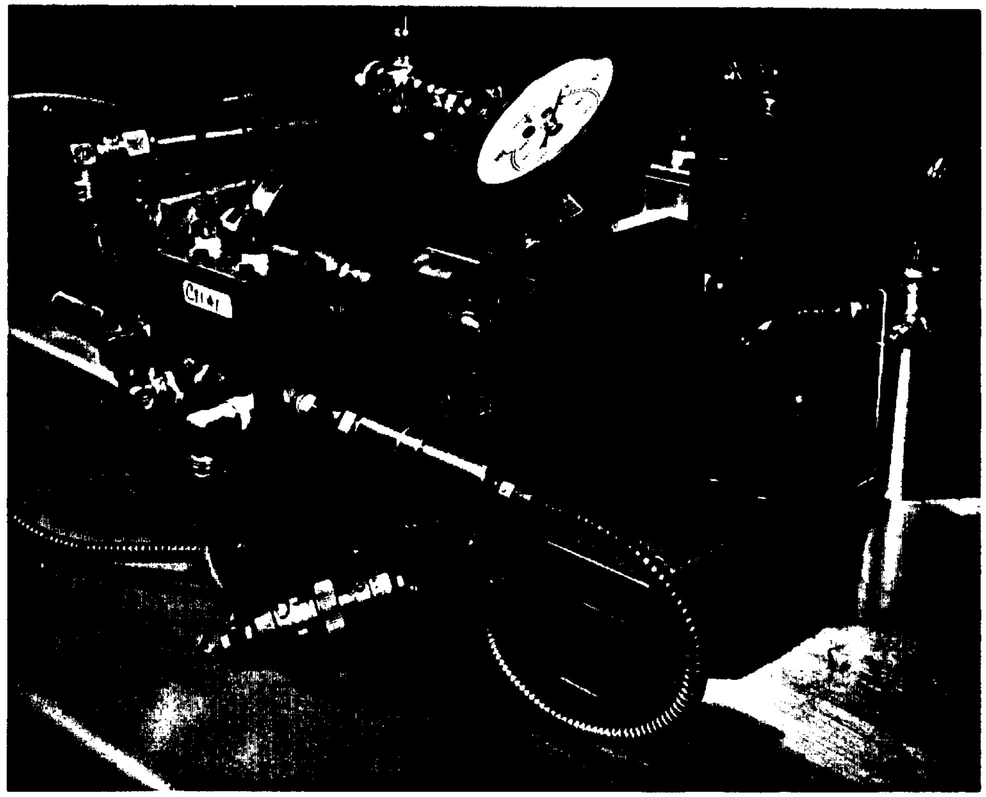


FIGURE 10. LABORATORY BREADBOARD TEST ARTICLE COMPRESSOR PROGRAM

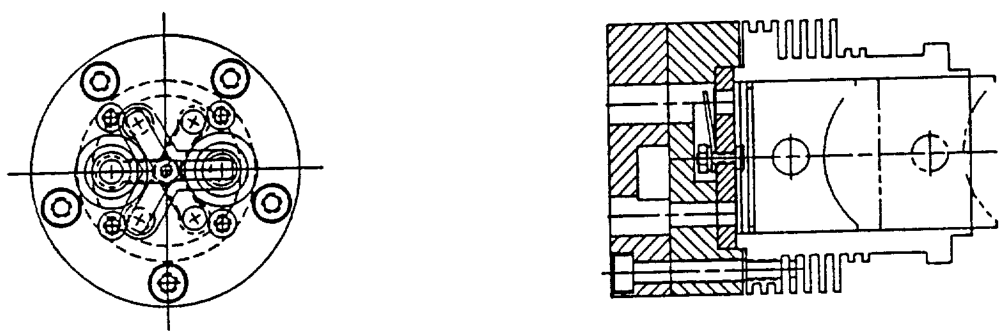
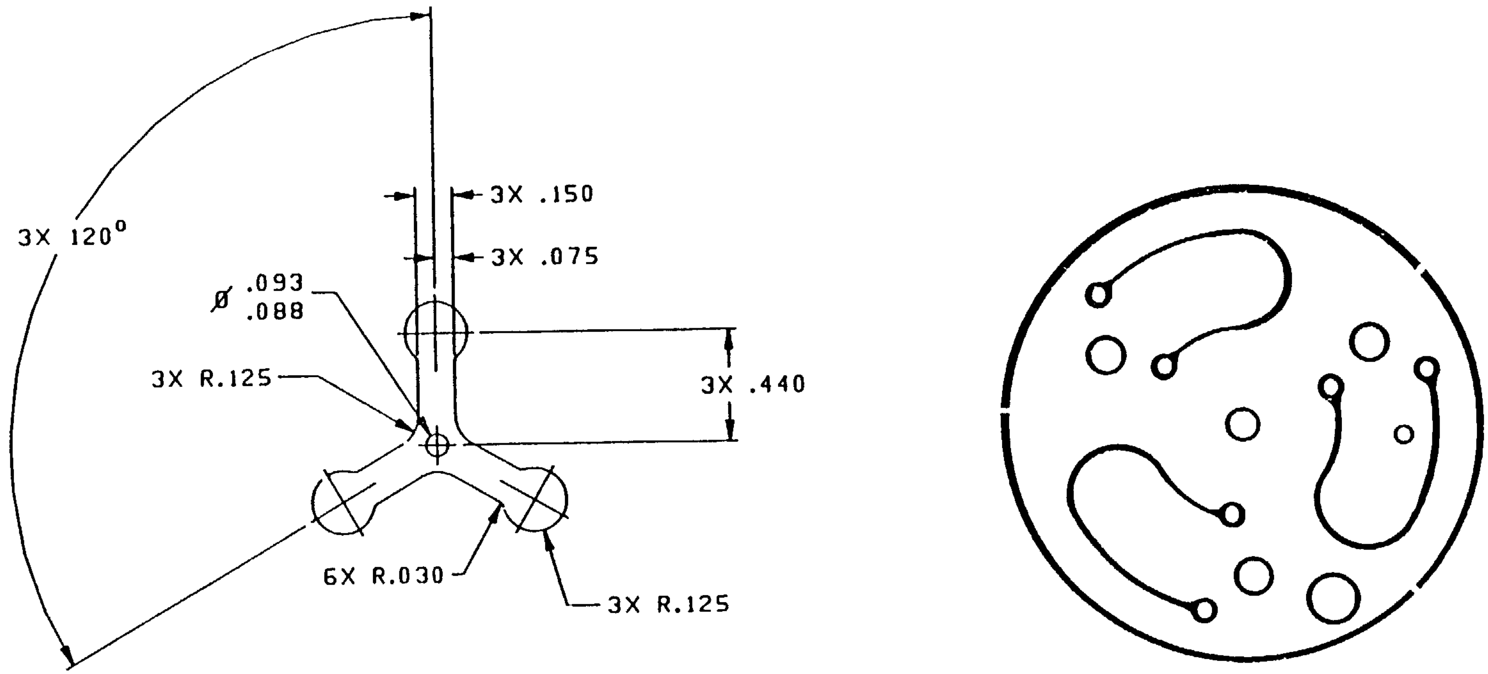


FIGURE 11. BREADBOARD TEST ARTICLE CYLINDER AND HEAD LAYOUT



MATL: FLAT SMOOTH STOCK EQUIVALENT TO SANDVIK GRADE 20C (CARBON STEEL) OR GRADE 7C27MO2 (STAINLESS)

THICKNESS: .004, .006, .008, .010

FIGURE 12. BTA REGULAR AND HIGH SWIRL VALVES

flow rate, in-cylinder pressure, and compressor speed. An automated analog to digital data acquisition system digitized the sensor signals at the trigger times provided by the encoder. The digitized data was then plotted on the computer screen in a parameter (pressure or temperature) versus crank angle form. The measured data could also be input to the digital simulation model for comparison to the predicted compressor performance at the same operating conditions.

5.1.3 Performance Testing Results

Tests were performed by first selecting the desired suction pressure, discharge pressure, coolant temperature and compressor speed. The compressor was then set at these conditions and allowed to come to thermal equilibrium. After the sensors were checked at the warmed condition, data was taken for several cycles and stored in a computer file. The file was then plotted on the computer screen and visually checked. Many different test runs were performed over the range of operating conditions outlined below:

Suction Pressure (psia)	10 - 15
Discharge Pressure (psia)	50 - 500
Pressure Ratio	4.7 - 40
Compressor Speed (RPM)	4 - 4000
Suction Temperature (°F)	70 - 200
Discharge Temperature (°F)	350 - 1200
Flow Rate (PPH)	0.4 - 2.8

A summary of the test pressure conditions are shown in Table VI.

The primary results of the compressor performance testing were improvements and validation of the compressor simulation model. The simulation model was then used to perform trade-off studies for the prototype compressor. The parameters studied included stroke, bore, number of stages, power requirements, and heat rejection rates. The simulations showed that by enhancing heat transfer in the cylinders, higher pressure ratios and fewer stages can be employed. Example P-V (Pressure-Volume) cards for the regular valve and high swirl valve configurations are shown in Figures 13 and 14, respectively. This validated model was then used as the design tool to size the prototype compressor.

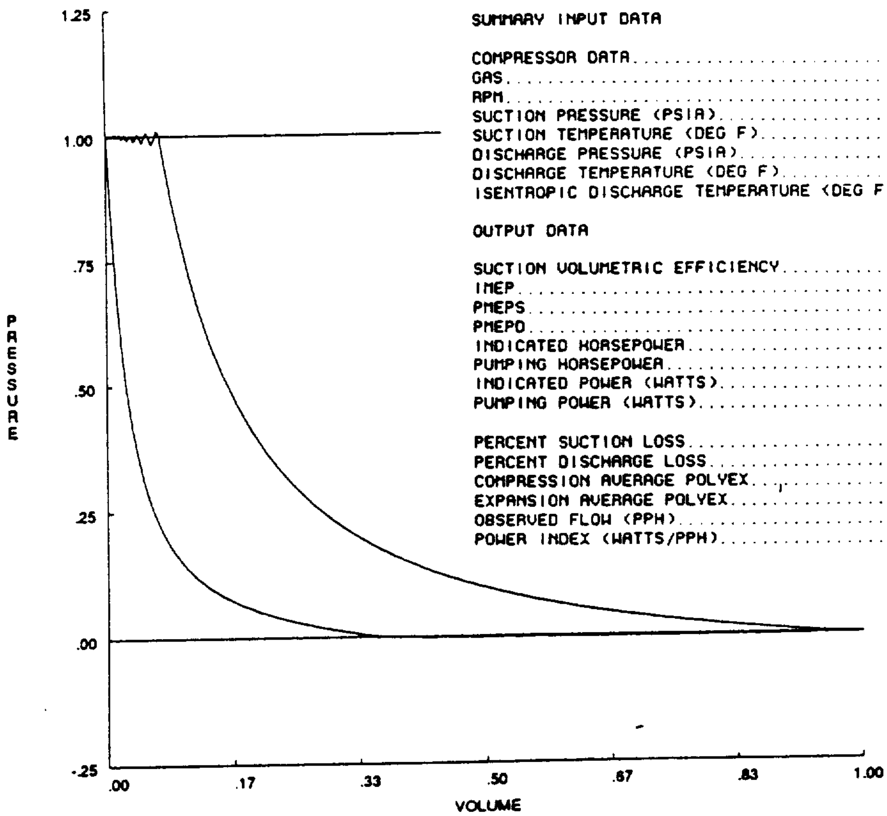
5.2 Subassembly Wear Testing

5.2.1 Wear Testing Objectives

The objective of this subassembly test program is to obtain wear data on candidate seal and guide ring materials for the prototype compressor. The data aided in the material selections for the

TABLE VI. SUMMARY OF BTA TESTS

File No.	RPM	P_S *(psia)*	P_D *(psia)*	P_R	T_S *(°F)*	T_{DO} *(°F)*	T_{DIS} *(°F)*	Flow *(PPH)*
5.4	500	15.33	217.4	14.18	72.7	92.9	681.9	.7803
5.5	500	14.54	113.8	7.83	117.3	174.2	583.1	1.0161
5.5	500	15.23	312.7	20.53	130.2	168.7	946.9	.5620
5.5	1000	15.51	307.1	19.80	159.6	263.7	1001.9	1.0161
5.5	1500	13.67	305.1	22.32	197.1	320.3	1144.7	1.4881
5.5	1000	11.09	524.1	47.26	92.4	147.5	1213.9	.4083
5.5	1000	10.51	336.9	32.05	91.1	178.6	1033.4	1.142
510	1500	11.43	179.5	15.70	99.5	253.4	775.1	2.163
513	500	14.11	116.7	8.27	69.6	116.3	512.2	1.026
513	750	13.04	118.8	9.11	75.9	154.5	551.5	1.512
513	1000	11.57	121.3	10.48	79.8	181.0	600.9	1.632
513	750	13.61	145.7	10.70	79.3	154.9	606.1	1.323
4.28	500	13.51	49.88	3.69	95.5	138.9	348.6	1.108
4.28	1000	12.59	50.3	4.0	115.3	201.2	396.7	2.029
4.28	1500	11.65	51.13	4.39	133.5	242.5	448.1	2.591
4.28	2000	10.40	52.65	5.06	150.3	293.0	512.8	2.784
4.28	500	14.64	70.51	4.81	101.8	154.8	422.7	1.038
4.28	500	14.48	91.84	6.34	110.7	171.8	510.7	.9534
4.28	500	14.40	111.24	7.72	117.6	183.5	579.6	.8987
4.28	1500	14.46	91.84	6.35	110.7	171.8	510.7	.9631
4.28	1500	12.21	90.93	7.48	162.1	312.8	648.15	2.2823
4.28	1500	12.39	114.48	9.24	172.7	335.5	739.0	2.1774
4.28	500	12.96	211.6	16.33	84.6	121.6	755.6	.6548
4.28	500	13.79	213.5	15.48	70.5	115.2	706.2	.6370
4.29	500	14.05	112.07	7.98	66.8	110.6	497.1	1.0175



SUMMARY INPUT DATA

COMPRESSOR DATA	NASA
GAS	AIR
RPM	499.0000
SUCTION PRESSURE (PSIA)	13.7891
SUCTION TEMPERATURE (DEG F)	70.5000
DISCHARGE PRESSURE (PSIA)	213.5000
DISCHARGE TEMPERATURE (DEG F)	115.2000
ISENTROPIC DISCHARGE TEMPERATURE (DEG F)	706.1936

OUTPUT DATA

SUCTION VOLUMETRIC EFFICIENCY	.6327
IMEP	36.5176
PMEPS	.2403
PMEPD	.0288
INDICATED HORSEPOWER	.0559
PUMPING HORSEPOWER	.0004
INDICATED POWER (WATTS)	41.7127
PUMPING POWER (WATTS)	.3073
PERCENT SUCTION LOSS	.6579
PERCENT DISCHARGE LOSS	.0788
COMPRESSION AVERAGE POLYEX	1.3085
EXPANSION AVERAGE POLYEX	1.3545
OBSERVED FLOW (PPH)	.6371
POWER INDEX (WATTS/PPH)	65.4729

FIGURE 13. TEST RESULTS FOR BTA COMPRESSOR WITH REGULAR VALVES

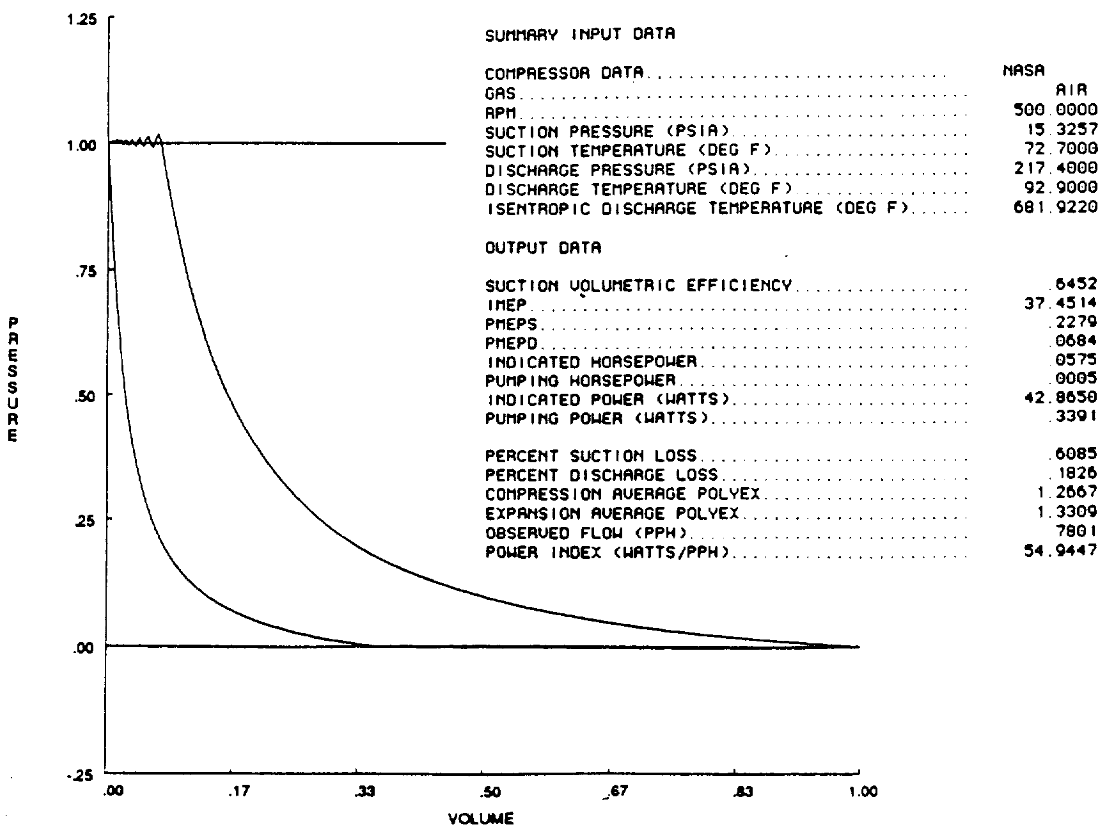


FIGURE 14. TEST RESULTS FOR BTA COMPRESSOR WITH HIGH SWIRL VALVES

prototype design and predicating compressor operating life. Although the wear test apparatus is primarily developed for seal and guide ring testing, information on the performance of the cam follower, actuator bearing and valves was also obtained.

5.2.2 Wear Test Apparatus

The tests were conducted on a single piston compressor designed similar to the prototype compressor. The piston is actuated by an eccentrically mounted bearing acting on an insert in the piston base. The piston is kept in contact with the bearing outer-race by a set of coil springs. Simple pressure actuated check valves are used on both the suction and discharge ports. The following table presents the subassembly test rigs specifications:

Cylinder Bore (inches)	1.125
Lower Guide Bore (inches)	2.282
Piston Stroke (inches)	0.65
Suction Pressure (psia)	Atmospheric
Discharge Pressure (psig)	100
Compressor Speed (RPM)	1000

This Subassembly Test Article (STA) was designed so the various seal and guide ring materials could easily be interchanged and new cylinder sleeves inserted in the compressor. Compressed air was circulated around the outside of the cylinder to keep the assembly cool. During wear testing the compressor speed, discharge pressure, and the discharge gas temperature (in the pulsation bottle) were monitored. During the course of testing, several different seal designs were evaluated so several different piston designs were required to accommodate the various seals.

5.2.3 Wear Testing Results

Prior to assembly of the STA, the seals, guide rings, and cylinder were weighed and measured. The compressor was then run and periodically disassembled for inspection and measurements. For several of the tests, simply inspecting the seals and the quantity of wear products was all that was necessary and then the compressor was reassembled and restarted. At the end of each test, the seals were remeasured to quantify the wear rate. The initial surface finish on the cylinder walls for all of the testing was 16 RMS. Seven different wear tests were performed. The materials for each are summarized in Table VII and the results are discussed below.

TABLE VII. WEAR TEST MATERIALS

TEST #	Reciprocating Seal	Guide Ring	Cylinder Material
1	O-ring energized Turcite-42 band, 0.072" Length	Turcite-42 band, 0.072" length	Clear anodized 6061-T6 aluminum
2	Spring energized graphite fiber reinforced PTFE with molybdenum disulfide "U" seal. Bal-Seal 415LB-212-GFPM	Torlon 4301 band, 0.25" length	Hardtuf X20 on 6061-T6 aluminum
3	Spring energized PTFE with molybdenum disulfide "U" seal. Variseal S32241-119-W-99S	Torlon 4301 band, 0.25" length	Hardtuf X20 on 6061-T6 aluminum
4	Variseal S32241-119-W-99S	Torlon 4301 band, 0.25" length	Hardtuf X20 on 6061-T6 aluminum
5	Torlon 4301 step cut seal run with and without expander ring. 0.120" length	Torlon 4301 band, 0.25" length	Hardtuf X20 on 6061-T6 aluminum
6	Hydlar-ZT step cut seal run with and without expander ring. 0.120" length	Torlon 4301 band, 0.25" long	Hardtuf X20 on 6061-T6 aluminum
7	Rulon-F step cut seal, with expander ring. 0.120" length	Rulon F band, 0.25" long	Hardtuf X20 on 6061-T6 aluminum

5.2.3.1 Test #1 Results

The compressor was run for 8 hours at 1000 RPM with the discharge pressure set at 120 psig. When the compressor was disassembled, a white powdery material covered the cylinder and valve assembly. The cylinder walls were lightly scoured below the seal ring path. The material removed from the cylinder was apparently the source of the powder. The cylinder wall opposite the guide ring showed material transfer from the guide ring and no apparent wear.

The seal was assembled following the manufacturer's guideline and appears to run too tight for non-lubricated applications. The clear anodized aluminum 6061-T6 material did not provide an adequate cylinder wall material. Because of concerns relating to O-ring set in the present seal design over long term, testing of metal spring energized seals was started. The cylinder was also changed to a Tiodize Hardtuf X20 coating on 6061-T6 cylinder.

5.2.3.2 Test #2 Results

The compressor piston was modified for a Bal-Seal seal ring and a Torlon 4301 guide ring and a Tiodize Hardtuf X20 cylinder was installed. The compressor was run for a total of 500 hours with the compressor stopped and the seals weighed at 100, 200, 433, and 500 hours. The compressor discharge pressure was 100 psig, suction pressure was atmospheric pressure, and the compressor speed was 1000 RPM. Because of the seal design, only weight change could be monitored (the soft seal could not easily be measured with a micrometer). The Torlon guide ring varied in both weight and size during the test as did a second Torlon guide ring (it was not run in the compressor, but was placed in a plastic bag and only removed during weight checks) used as a weight check. The variations were caused by water absorption.

Each time the compressor was disassembled, the wear products from the seal were visible as dust on the cylinder and valves. The compressor ran fine for the duration of the test and was shut down to try other materials. The design of this commercially available seal does not allow much wear since it has only a thin amount of material covering the canted coil spring and limited spring travel. The material combination of "GFPM" and Tiodize X20 seemed to work well. No wear could be measured on the cylinder. The following table summarizes the weight loss of the seal during the test (seal start weight was 1.445 gm). As indicated above, the Torlon weight and size varied during the test (probably due to water absorption) so no detailed information is available on the Torlon wear. From a qualitative point of view, the surface of the Torlon appeared polished and no significant wear was evident.

Time (hours)	Cumulative Weight Loss (gm)
100	0.0060
200	0.0091
433	0.0149
500	0.0176

5.2.3.3 Test #3 Results

The same test conditions as above were run with a finger spring energized PTFE "U" seal that contained molybdenum disulfide lubricant. The same cylinder was used. The test only ran about one hour before maximum pressure dropped to 40 psig. The compressor was disassembled and the seal showed severe wear and the test was stopped.

5.2.3.4 Test #4 Results

Test #3 was repeated using a new seal and allowing an 8-hour break-in period with no back pressure on the compressor. After the 8-hour break-in period, the compressor was disassembled

and the seal inspected. The seal had again worn severely (for such a short test) and a considerable amount of wear products were on the cylinder and valve plate. The test was stopped at this point to try a seal design that has the potential for longer life. The unfilled PTFE appears to be too soft for use in an unlubricated environment.

5.2.3.5 Test #5 Results

The compressor piston was again modified so a step cut seal ring could be used. This test used a Torlon 4301 seal ring and guide ring. The initial testing was done without an expander ring under the seal ring. After 69 hours of testing, an expander spring was installed. The total test duration was 282 hours with the compressor shut down and inspected at the 69, 116, 188, and 282 hour points. The wear on the Torlon was even and only a relatively small amount of wear products accumulated in the cylinder.

The Torlon wear was excessive for such a short duration test. It is difficult to determine a wear rate for this test because of the changes during testing and the relatively large uncertainty in the small wear measurements. The post-break-in wear rate with the expander ring was about 0.032"/1,000 hours and without the expander 0.034"/1,000 hours.

5.2.3.6 Test #6 Results

Test #5 was repeated (same seal design and cylinder) with the seal made of Hydlar-ZT material. The test was run for a total of 133 hours. At the 42 hour mark, an expander ring was installed under the Hydlar seal ring. At this point there was fine dust particles on the valve plate, cylinder and piston. By hour 62, the maximum pressure (with flow valved off) was 75 psig so the compressor was disassembled and inspected. It is believed the wear particles may have lodged under the valve seats and caused the valves to leak. The compressor was reassembled and restarted after some minor modifications. The compressor finished the duration of the test with the discharge pressure at 100 psig. The total radial wear on the ring was 0.005 inches over 133 hours (for a wear rate of 0.038"/1,000 hours).

5.2.3.7 Test #7 Results

This test was performed with a Rulon F seal (same design as Test 5 and 6) and Rulon F guide ring. A new cylinder sleeve was installed in the compressor because the old one was showing score marks from the previous tests. The new cylinder was also made of aluminum with a Tiodize "Hardtuf X20" coating. An expander ring was also installed under the seal. The seal was run for a 1 hour break-in period with no back pressure and then the back pressure was increased to 100 psig. The following table presents the seal wear data over the 1800 hour test. The cylinder inside

diameter increased between 0.0006 to 0.0014 inches over the duration of the test. The wear was greatest in the plane of the actuator bearing because of the radial loads imparted by the piston actuator.

The following table gives the measured seal thickness during the testing. As with the other tests, the cylinder bore diameter was 1.125 inches and this seal contact length with the cylinder was 0.120 inches.

Test Time (Hours)	Seal Thickness (Inches)	Cumulative Loss (Inches)
0	0.0770	0
26	0.0767	0.0003
212	0.0745	0.0025
378	0.0741	0.0029
544	0.0737	0.0033
879	0.0715	0.0055
1213	0.0694	0.0076
1836	0.0664 to 0.0685	0.0106 to 0.0085

The seal thickness data presented is an average of five measurements evenly spread around the circumference. The last data point, however, exhibited a "flat-spot". Upon examination of the test rig the lower guide ring had worn through and the piston was not centered. This failure of the test rig occurred sometime between the 1200 hour point and the 1800 hour point. The cumulative loss from the 1800 hour point is somewhere in the range of 0.0106 inches (average for all five locations) and 0.0085 inches (average for four locations by eliminating the flat-spot). If the test rig failure had not occurred it is our judgement that the actual loss would be between the "worst-case" condition and optimum condition.

5.2.3.8 STA Test Observations

In addition to the above information gathered on the seals and guide rings, inspection of the cam follower, actuator bearing outer race and valves provided the following:

> ***Cam Follower:*** The cam follower was made of 4340 hardened to 50 Rc and then ground flat. Throughout the duration of all the wear testing, the cam follower showed no visible signs of wear or failure. Based on the first few tests, it appears the piston does not rotate (there is nothing to prevent the piston from spinning in the cylinder) from the position it is inserted in the cylinder. The contact marks on the follower showed a single strip of contact and not a circular region.

Actuator Bearing: No significant wear of the bearing outer race in contact with the cam follower was visible. No sliding contact on the bearing outer race (due to slipping of the cam follower over the race) was evident. No apparent problems, noise or vibration were evident from the bearing during the testing.

Check Valves: The compressor check valves suffered no failures during testing, but some performance degradation was noted during testing that was attributed to leakage in the valves. The wear test apparatus used a head assembly (containing the valves) from the breadboard performance testing. There were 3 suction valves and 3 discharge valves, each 0.188 inches in diameter in the head assembly. During testing, it appears some of the seal wear products would accumulate in the valve seats resulting in valve leakage and a decrease in compressor flow or outlet pressure. By plugging two suction and two discharge valves, the problem either went away or became less noticeable. Whether the increased gas velocity through the remaining valves transported the wear products away or there was less area for leakage is not clear. Slight leakage was also noticed between the ground and lapped (no elastomer face seals) valve plates.

6.0 DETAILED DESIGN DRAWINGS

Detailed design drawings are contained in Appendix A for each of the compressor components. The following is a list of the drawing numbers and drawing titles that are contained in the appendix.

DRAWING NUMBER	DRAWING NAME
2529001	Design Layout, Type II Mixed Gas Compressor
2529002	Flange, Front
2529003	Spacer, Bearing
2529004	Retainer, Motor
2529005	Washer, Motor
2529006	Flange Assembly
2529007	Crankshaft Balance Information Sheet
2529008	Retainer, Front Bearing
2529009	Retainer, Rear Bearing
2529010	Spacer, Crankshaft
2529011	Bearing Support, Front
2529012	Crankshaft Assembly
2525013	Bearing Support, Rear
2529014	Main Shaft
2529015	Housing Assembly
2529016	Feedthrough, Electrical
2529017	Nut, Feedthrough

DRAWING NUMBER	DRAWING NAME
2529019	Cylinder, Second Stage
2525020	Piston, First Stage
2529021	Piston, Second Stage
2529022	Follower, First Stage Piston
2529023	Follower, Second Stage Piston
2529024	Drawing List
2529025	Seat, Piston Spring
2529026	Pin, Piston Locating
2529027	Retainer, First Stage Outlet Valve
2529028	Retainer, First Stage Outlet Valve
2529029	Nut, First Stage Inlet Valve Retainer
2529030	Retainer, Second Stage Outlet Valve
2529031	Retainer, Second Stage Inlet Valve
2529032	Valve Plate, First Stage
2529033	Valve Plate, Second Stage
2529034	Manifold, Cylinder Cooling Fluid
2529035	Pulsation Chamber Assembly (Sheet 1)
2529035	Pulsation Chamber Assembly (Sheet 2)
2529036	Piston Ring, First Stage
2529037	Piston Ring, Second Stage
2529038	Cooling Plate, Motor

7.0 PROCUREMENT SPECIFICATIONS

7.1 Piston Return Spring

The piston return spring provides the force for drawing gas into the cylinder during the suction portion of the cycle and keeps the cam follower in contact with the actuator. Both the first and second stages use the same return springs. The spring is designed to survive 10^7 cycles with a 15 LB preload, 0.48 inch stroke and 60 LB/inch spring rate. The following specifications define the spring geometry and materials.

Wire Diameter (inches)	0.135
Inside Diameter (inches)	0.890 ±0.015
Free Length (inches)	2.20 ±0.03
Spring Rate (LB/in)	60
Total Coils	9.3
Active Coils	7.3
Helix Direction	Right Hand
Shot-peened	YES
Material	302SS
End Closed and Ground	YES
Squareness (degrees)	3
Parallelism (degrees)	3

7.2 Static Seals

Static face seals are used in the compressor for sealing the case, cylinders, pulsation bottles and the cooling jacket. Spring energized "C" ring seals were selected to provide long life, reliable seals. The 301 stainless steel spring provides compatibility with the gas mixture and is not as prone to creep, set, or thermal degradation as an all elastomer seal. The material selected for the seal itself is PTFE. This inert material provides excellent compatibility with the gas mixture, long shelf life, and is rated for use from -320°F to 450°F.

7.3 Drive Motor and Controller

The drive motor in the prototype compressor is a high torque brushless DC motor. The motor was selected because it provides high torque and low weight in a small package. The brushless design will also provide a long service life and will not produce wear products associated with brush wear. The performance specification and dimensional information on the motor and controller are contained in the manufacturer's literature reproduced in Appendix B. The motor and controller part numbers are Inland Motor RBE-01804-X00 and BLM1-02820H0X, respectively. The power supply required for the motor controller is 28 VDC at 20 ADC.

It should be noted that this motor and controller are not being tested as flight hardware, but rather to demonstrate them as generic devices capable of properly powering the compressor.

7.4 Fasteners

All nuts, bolts, and washers used in the compressor are made of 304SS. All of the threads taped into the aluminum case and motor housing have 302SS Heli Coil self-locking threaded inserts. These inserts strengthen the taped threads by uniformly distributing the loading and also increase thread life. The threads on the crankshaft ends and motor electrical feedthrough were self-locking spiral lock threads.

8.0 FABRICATION NOTES

Material specifications and fabrication notes are contained in the detailed drawings in Appendix A.

9.0 PROPOSED MODIFICATIONS TO END ITEM SPECIFICATION

In general, the EIS represents a good specification for flight hardware. There are a few sections that can be improved for the specific application of an on-orbit compressor. The sections that should be modified are: Surface Wear (3.2.2.3), Lubricants (3.3.1.5), Performance (4.2.2 & 4.2.4), Proof Pressure (4.2.2.2), and Service Life (4.2.5).

9.1 Surface Wear

The current EIS states that, ". . . shall not introduce contaminant into the fluid flow path . . .". Each compressor type has a different application, and the effect of wear particles is different. Specifically, a realistic acceptable number and size of wear particle for the waste gas compressors should be stated.

9.2 Lubricants

The current EIS states that, ". . . do not introduce contamination by entering the fluid flow path." As indicated above for wear particles, a realistic acceptable level of lubricant transfer downstream for the waste gas compressor should be stated.

9.3 Performance

The current EIS provides inlet and outlet pressure ranges and a flow rate range independent of each other. Since the flow rate is not independent of inlet and outlet pressure, specific combined operating conditions should be stated. As an example, at an inlet (suction) pressure of 10 psia and an outlet (discharge) pressure of 1000 psia, the fluid flow rate shall be 0.25 lbm/hr. The performance of a compressor is best illustrated in the form of a discharge pressure versus fluid flow rate curve at a given inlet pressure and rotational speed. The performance curve can be specified by three points: the pressure at zero flow rate (i.e., deadhead pressure), the flow rate at zero pressure rise (i.e., flow rate when suction pressure equal to discharge pressure), and a nominal flow rate at a nominal pressure. This approach to specifying performance assumes a constant compressor rotational speed. Each rotational speed will have a different curve with a different deadhead pressure (i.e., maximum pressure) at no flow and zero-pressure-rise flow rate (i.e., maximum flow rate at no pressure rise). The control strategy for motor speed is also important. A constant speed compressor greatly simplifies the control system, but a variable speed system provides more flexibility in pressure versus flow rate combinations. Since the waste gas application is to pump up a reservoir from 100 psia to 1000 psia, the important issue is the flow rate. At a constant speed, the flow rate will be high at 100 psia, i.e., the beginning of the cycle and gradually decline as the vessel pressure approaches 1000 psia. If a constant flow rate is required over the entire range of discharge pressures, then a variable speed is required. This capability will result in a more complex control system and a larger capacity unit. Once you have determined how you intend to operate the unit, then a more specific performance specification can be written with the above guidelines in mind.

9.4 Proof Pressure

The current EIS requires that the entire compressor be subjected to a proof pressure of 1.5 times the maximum discharge pressure for five minutes and designed for a burst pressure of 2.5 times the maximum discharge pressure. Since the case is vented to suction pressure, the requirement that the case withstand this proof pressure results in a significantly heavier case than if the case proof pressure was 1.5 times the highest pressure it would experience (i.e., suction pressure). To be more specific, the design burst pressure for the case is 2.5 times the maximum discharge pressure or 2500 psia. However, this is a hundred times the maximum suction pressure. A more realistic requirement would greatly reduce the weight of the case and total weight of the compressor.

9.5 Service Life

The current EIS requires 10,000 operating hours of continuous duty. This life requirement with an unlubricated compressor is severely pushing the state-of-the-art (SOA). As the program proceeded, the objective of maximizing life potentially up to 10 years (876,000 hours) was recommended which is beyond the SOA. Two specific modifications are recommended for the waste gas compressor. The first is to identify a realistic duty cycle, and the second is to allow a lubricated unit. A realistic duty cycle can result in significant factors of life extension, i.e., if the compressor is realistically only on 1/4 of the time the life can be extended by a factor of 4 for dry seals. For lubricated seals, the life prediction is more complex because wear is not only a factor of operating life but also the number of starts. The life prediction for a lubricated unit, also, has the complexity of the life of the lubricant. However, even with these added complexities in life prediction, the life of lubricated units is significantly longer than unlubricated units. The specification should be modified to include operating hours and duty cycle.

10.0 FAILURE MODES AND EFFECTS ANALYSIS

The Failure Modes and Effects Analysis (FMEA) serves to identify possible compressor failure modes, failure causes, and the effect each failure mode has on the system operation. Once the failure modes and effects are defined, they can be used to guide design decisions, safety analyses, and hardware test and inspection plans.

The process of conducting the FMEA consists of analyzing each hardware item for each possible failure mode and for the "worst case" effects of the failures. The analysis includes the interrelationships between control systems, operating environments, external interfaces, and the operating hardware. By looking at the interrelationships, the effect of a component failure on the overall system operation can be determined.

The FMEA in the following tables contains the item name, functional description, failure mode and cause, failure effects, failure detection, and corrective action required.

TABLE VIII. FAILURE MODES AND EFFECTS ANALYSIS

Part Name	Function	Failure Mode and Cause	Failure Effect On System	Failure Detection	Corrective Action	Time To Effect*	Rev
A. Roller Bearings:	Provide low friction crankshaft support.	Bearing seizure due to wear and pitting. a) Overload due to other failures (misalignment, piston sticking, broken return spring) or discharge pressure rise. b) Loss of bearing lubricant causing overheating and scoring.	Compressor stops. Noisy operation, increased vibration, increased power requirement, and crankcase contamination with lubricant and metal particles.	Power consumption monitor, vibration sensors, speed sensor.	Replace compressor	Immediate	
B. Actuator Bearings:	Provide low friction interface between cam follower and crankshaft.	Bearing seizure due to wear and pitting. a) Overload due to other failures (misalignment, piston sticking, broken return spring) or discharge pressure rise. b) Loss of bearing lubricant causing overheating and scoring.	Compressor stops. Noisy operation, increased vibration, increased power requirement, and crankcase contamination with lubricant and metal particles. Increased friction between bearing and cam follower.	Power consumption monitor, vibration sensors, speed sensor.	Replace compressor	Immediate	

* Slow > 12 hours, Moderate = 5 min. to 12 hours, Immediate = 0 - 5 minutes.

C-2

Part Name	Function	Failure Mode and Cause	Failure Effect On System	Failure Detection	Corrective Action	Time To Effect*	Rev
C. Gas Check Valves:	Control gas flow through suction and discharge ports.	Check valves do not provide sealing. a) Fatigue of valve causing cracking and breakage. Possibly corrosion assisted fatigue. b) Debris lodging on seat will not allow valve to seal properly. c) Wear or corrosion on seat surface allowing valve leakage.	Poor compressor efficiency. Decrease in maximum discharge pressure.	Maximum output pressure reduced. Pressure sensors on innerstage and discharge stage would show changing pressure ratios (one stage doing all the compressing).	Replace compressor	Slow	
D. Piston Seals:	Provide seal between piston and cylinder wall.	Failure to seal gas in cylinder above piston. a) Wear of seal material beyond limit of seal energizer spring travel. b) Chemical attack on seals causing changes in wear and elasticity properties. c) Failure of energizer spring due to chemical attack or to taking a "set."	Poor compressor efficiency. Decrease in maximum discharge pressure.	Maximum output pressure reduced. Pressure sensors on innerstage and discharge stage would show changing pressure ratios (or stage doing all the compressing).	Replace compressor	Slow	

Part Name	Function	Failure Mode and Cause	Failure Effect On System	Failure Detection	Corrective Action	Time To Effect*	Rev
E. Wear/Support Rings:	Provide lateral piston support.	Failure to support piston. a) Excess wear on support rings allowing piston too much lateral movement. b) Chemical attack resulting in material property changes.	Initially increased piston vibration in cylinder eventually leading to piston sticking.	Vibration sensors to detect piston vibration in cylinder.	Replace compressor	Slow	
F. Piston return spring:	Keeps cam follower in contact with cam and provides gas suction into cylinder.	Spring force reduction or spring breakage. a) Spring material fatigue and fracture. b) Contamination induced corrosion accelerate fatigue.	Loss in compressor performance. Increased vibration. Piston will not follow cam. Piston will knock into head and follower.	Innerstage and discharge pressure monitors, vibration sensors.	Replace compressor	Immediate	
G. Cam follower:	Provides the interface between piston and the actuator bearing on the crankshaft.	Surface wear and pitting. a) High contact stress due to piston sticking. b) Impacting due to piston sticking or spring failure. c) Material property changes due to temperature or contamination.	Reduced compressor efficiency (wear of follower will cause increased cylinder clearance volume).	Discharge pressure sensor.	Replace compressor	Slow	

Part Name	Function	Failure Mode and Cause	Failure Effect On System	Failure Detection	Corrective Action	Time To Effect*	Rev
H. Static seals:	Provide gas seal in piston head and case seals. Provide seals for cooling fluids.	Leakage past seal. a) Chemical attack of seal by contaminants resulting is altered material properties. b) Cyclical loading causing seal movement in groove and associated wear. c) Material property change due to over temperature event.	Leakage of gas to atmosphere or between suction and discharge parts. Cooling fluid leakage into gas stream or to atmosphere.	Coolant quantity gage, innerstage and discharge pressure sensors, external leak detectors.	Isolate compressor and refurbish.	Slow.	
I. Coolant Passages	Provides heat removal from head, cylinders, and motor.	Motor or compressor overheating. a) Coolant pump failure. b) Cooling fluid passage blocked. c) Coolant leak.	Compressor shutdown due to overheating of motor or cylinders.	Temperature sensor on motor and head.	Isolate and repair cooling system. Replace compressor, or refurbish compressor.	Moderate	
J. Crankshaft:	Provides piston actuation and motor rotor support.	Crankshaft deformation and breakage. a) Fatigue due to overloading by dragging piston or increased discharge pressure. b) Material corrosion accelerating fatigue. c) Material flaws in crankshaft.	Compressor stops or vibration increases if crank is deformed.	Vibration sensors and power monitors.	Replace compressor.	Immediate	

Part Name	Function	Failure Mode and Cause	Failure Effect On System	Failure Detection	Corrective Action	Time To Effect*	Rev
K. Compressor case:	Provides housing and support for motor, bearings, and pistons.	Warping or rupture. a) Overpressure of case by leaking from piston or discharge line and blockage of case vent line. b) Thermally induced warpage. c) Material flaws. d) Damage in handling.	Compressor seizing due to misalignment. Gas and coolant leakage internal and external to case.	Pressure sensors, leak detectors, speed sensors.	Replace compressor	Slow	

APPENDIX A - DETAILED FABRICATION DRAWING PACKAGE

DWG NO		DWG SIZE		DWG TITLE		FILE NAME
2529001	-	D	-	DESIGN LAYOUT, TYPE II MIXED GAS COMPRESSOR	-	(COMPR.NAS)
2529002	-	D	-	FLANGE, FRONT	-	(FLGFR.NAS)
[illegible]			-	SPACER, BEARING	-	[illegible].NAS)
2529004	-	D	-	RETAINER, MOTOR	-	(RETMTR.NAS)
2529005	-	B	-	WASHER, MOTOR	-	(WASHMTR.NAS)
2529006	-	D	-	FLANGE ASSEMBLY	-	(FLGASYRR.NAS)
2529007	-	D	-	CRANKSHAFT BALANCE INFORMATION SHEET	-	(CRKBAL.NAS)
2529008	-	D	-	RETAINER, FRONT BEARING	-	(RTBGFT.NAS)
2529009	-	D	-	RETAINER, REAR BEARING	-	(RTBGRR.NAS)
2529010	-	D	-	SPACER, CRANKSHAFT	-	(SPCRCRK.NAS)
2529011	-	D	-	BEARING SUPPORT, FRONT	-	(BGSPFR.NAS)
2529012	-	D	-	CRANKSHAFT ASSEMBLY	-	(CRKASSY.NAS)
2529013	-	D	-	BEARING SUPPORT, REAR	-	(BGSPRR.NAS)
2529014	-	D	-	MAIN SHAFT	-	(MAINSH.NAS)
2529015	-	D	-	HOUSING ASSEMBLY	-	(HOUSASSY.NAS)
2529016	-	D	-	FEEDTHROUGH, ELECTRICAL	-	(FDTHRU.NAS)
2529017	-	B	-	NUT, FEEDTHROUGH	-	(NUTFDTH.NAS)
2529018	-	D	-	CYLINDER, FIRST STAGE	-	(CYLFRST.NAS)
2529019	-	D	-	CYLINDER, SECOND STAGE	-	(CYLSEC.NAS)
2529020	-	D	-	PISTON, FIRST STAGE	-	(PISTFRST.NAS)
2529021	-	D	-	PISTON, SECOND STAGE	-	(PISTSEC.NAS)
2529022	-	B	-	FOLLOWER, FIRST STAGE PISTON	-	(FOLFRST.NAS)
2529023	-	B	-	FOLLOWER, SECOND STAGE PISTON	-	(FOLSEC.NAS)
2529024	-	B	-	DRAWING LIST	-	(DRAWLIST.NAS)
2529025	-	B	-	SEAT, PISTON SPRING	-	(SEATSPG.NAS)
2529026	-	B	-	PIN, PISTON LOCATING	-	(PINLOC.NAS)
2529027	-	B	-	RETAINER, FIRST STAGE INTAKE VALVE	-	(RETVALII.NAS)
2529028	-	B	-	RETAINER, FIRST STAGE OUTLET VALVE	-	(RETVALIO.NAS)
2529029	-	B	-	NUT, FIRST STAGE INLET VALVE RETAINER	-	(NUTVALRE.NAS)
2529030	-	B	-	RETAINER, SECOND STAGE OUTLET VALVE	-	(RETVLIIO.NAS)
2529031	-	B	-	RETAINER, SECOND STAGE INLET VALVE	-	(RETVLIII.NAS)
2529032	-	D	-	VALVE PLATE, FIRST STAGE	-	(VALPLAI.NAS)
2529033	-	D	-	VALVE PLATE, SECOND STAGE	-	(VALPLAII.NAS)
2529034	-	D	-	MANIFOLD, CYLINDER COOLING FLUID	-	(MANFCOOL.NAS)
2529035	-	D	-	PULSATION CHAMBER ASSEMBLY (SHEET 1)	-	(CHAPUL.NAS)
2529035	-	D	-	PULSATION CHAMBER ASSEMBLY (SHEET 2)	-	(CHAPULII.NAS)
2529036	-	B	-	PISTON RING, FIRST STAGE	-	(RINGI.NAS)
2529037	-	B	-	PISTON RING, SECOND STAGE	-	(RINGII.NAS)
2529038	-	D	-	COOLING PLATE, MOTOR	-	(MOTCOOL.NAS)

PARTS LIST

UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	NOTE TEXT (STOCK,SIZE,MATERIAL,SPECIFICATION,TYPE,CLASS,GRADE,HEAT TREAT)

APPLICATION

NEXT ASSY.	USED ON

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES AND INCLUDE CHEMICALLY APPLIED OR PLATED FINISHES

TOLERANCES

BASIC DIMENSION	DECIMALS 2 PLACE	DECIMALS 3 PLACE	FRACTIONS
UNDER 6	±.01	±.005	±1/32
6-24 INCL.	±.03	±.010	±1/16
OVER 24	±.06	±.015	±1/8
ANGLES	± 0°-30'		

COML TOL APPLY TO STOCK SIZES

CONTRACT		
DRN	E. C. SCHROEDER	4-23-90
CHK	E. C. SCHROEDER	9-25-90
MECH.		
ELEC.		
PROJ.		

SOUTHWEST RESEARCH INSTITUTE

DRAWING LIST

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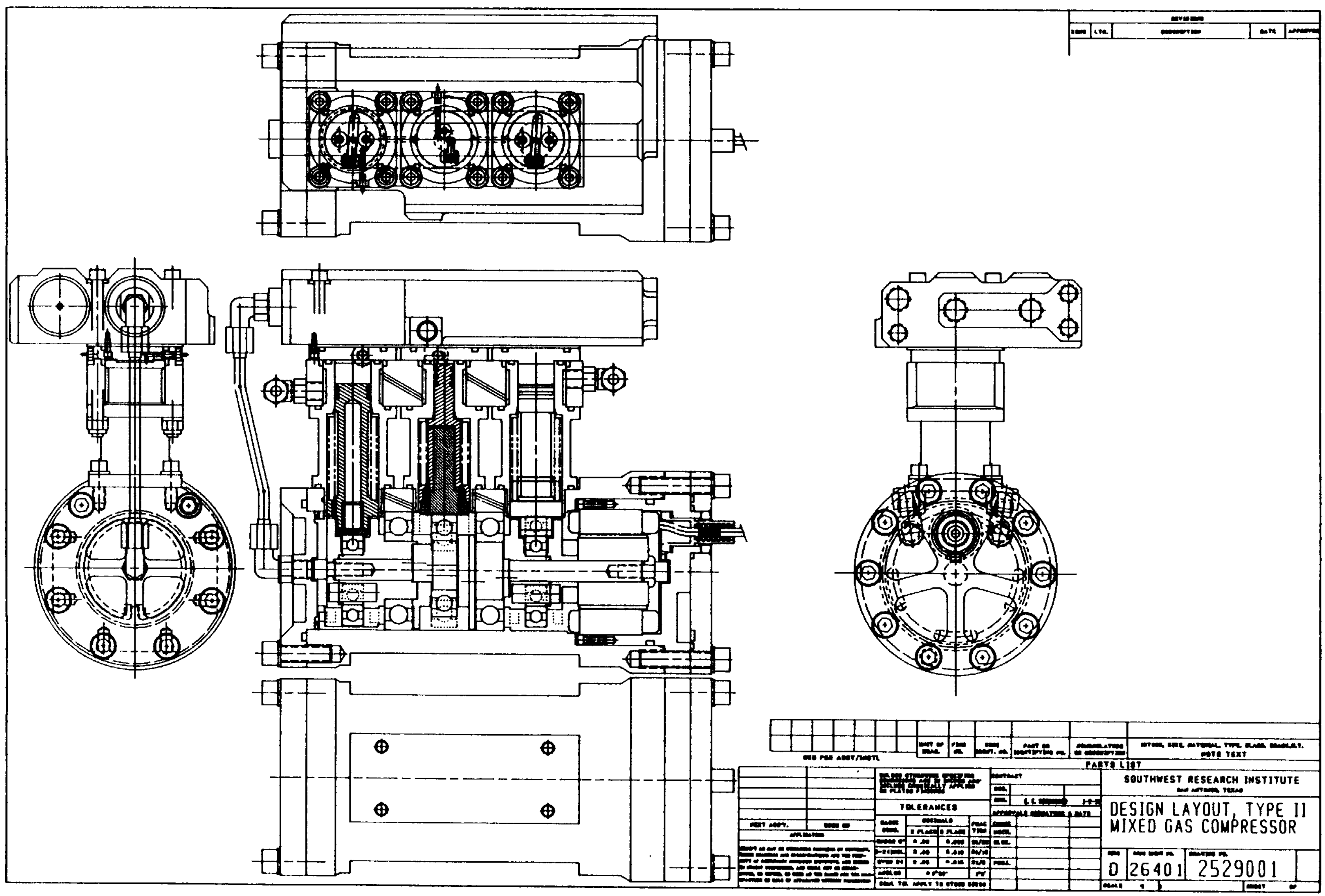
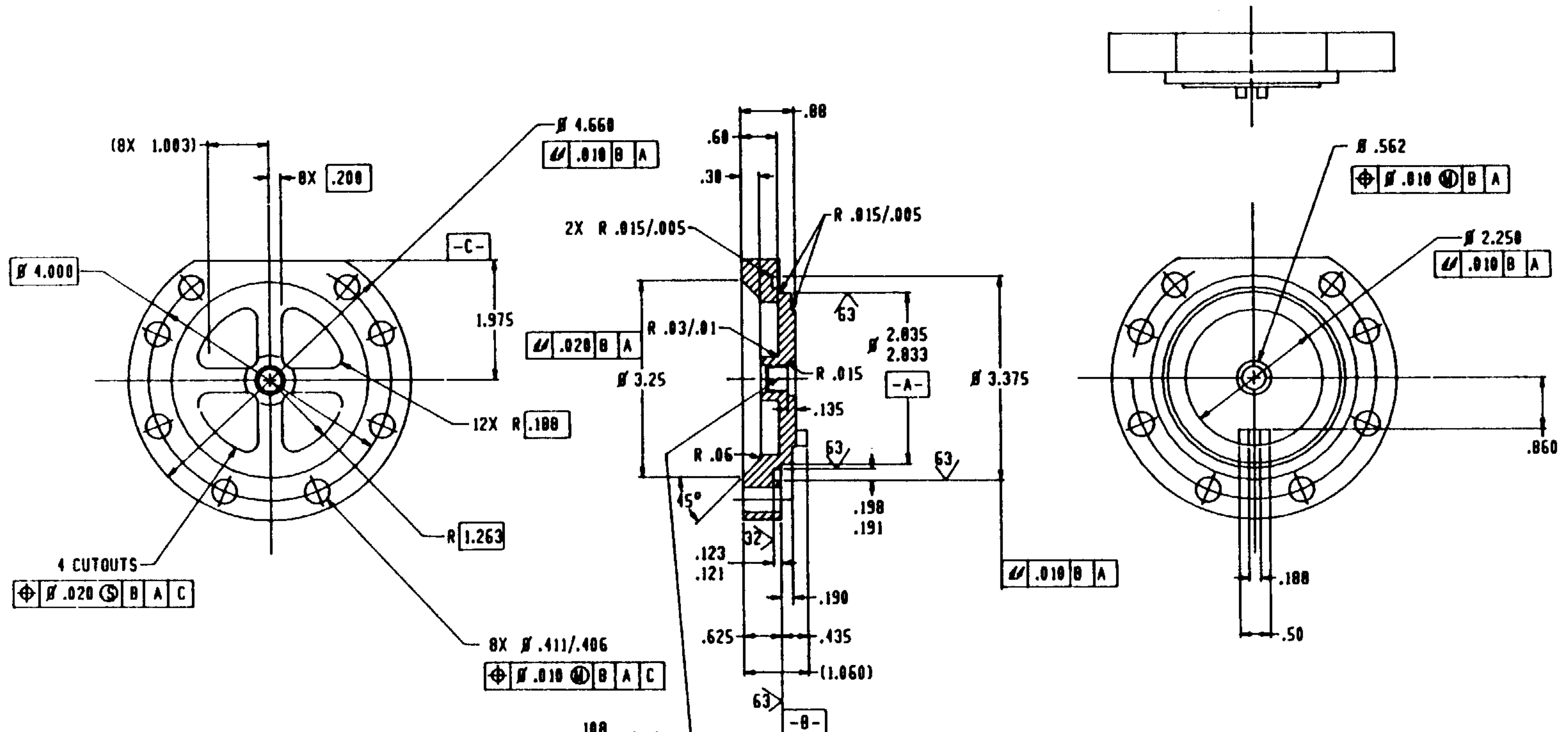
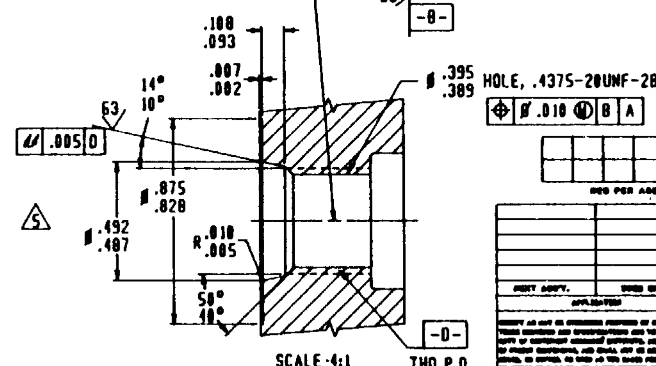


REVISIONS
PARTS LIST
TOLERANCES
SOUTHWEST RESEARCH INSTITUTE
SAN ANTONIO, TEXAS
DESIGN LAYOUT, TYPE II
MIXED GAS COMPRESSOR
D
26401
2529001





NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND EDGES EQUIV TO R .005
3. SURFACE FINISH 125 RMS UNLESS SHOWN
4. ANODIZE PER MIL-A-8625C, TYPE 2, CLASS 1
5. PORT CONTOUR MAY BE MACHINED USING A SIZE 4 SAE FORM TOOL (MS16142). PARKER HANNIFIN CORP PART NO Y34732.

-1		ALUMINUM, 6061-T6

PARTS LIST

TOLERANCES

SOUTHWEST RESEARCH INSTITUTE

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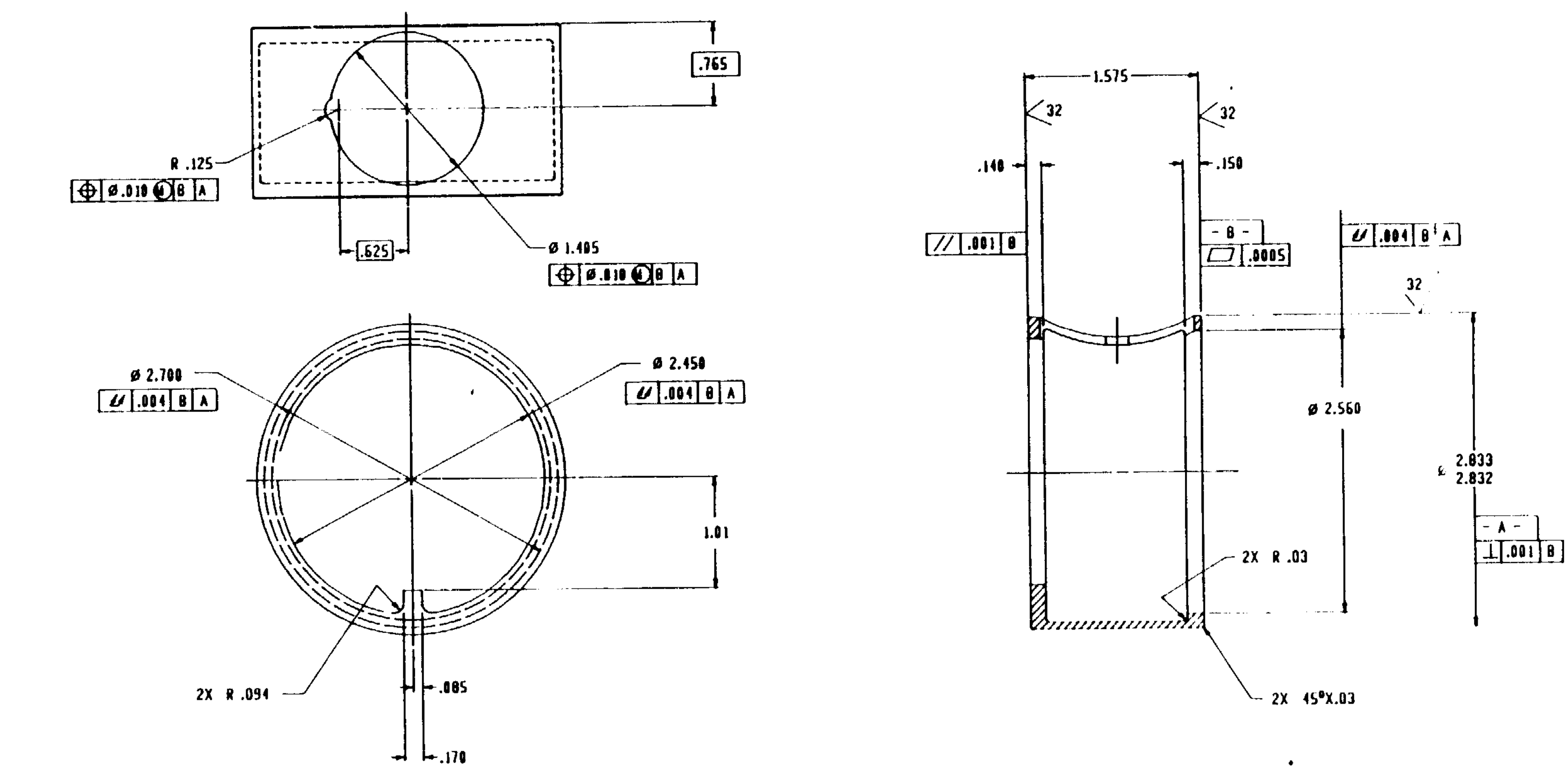

NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND SHARP EDGES EQUIV TO R.005
3. SURFACE FINISH 125 RMS UNLESS SHOWN
4. ANODIZE PER MIL-STD-8625C, TYPE 2, CLASS 1

AL, 6061-T6

PARTS LIST

TOLERANCES

SOUTHWEST RESEARCH INSTITUTE

SAN ANTONIO, TEXAS

SPACER, BEARING

D 26401 2529003

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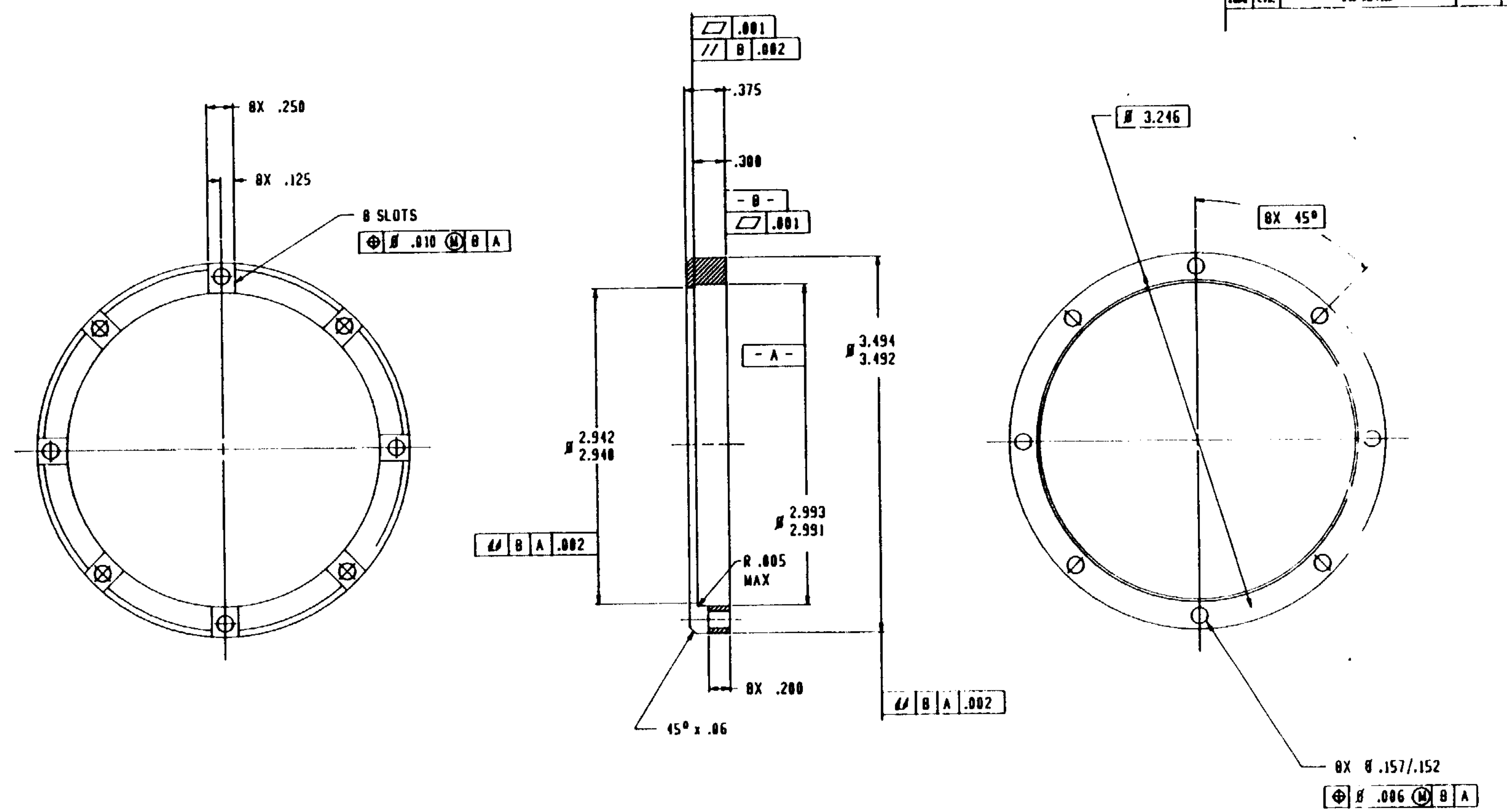

NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND EDGES EQUIV TO R.005
3. SURFACE FINISH 63RMS UNLESS SHOWN
4. ANODIZE PER MIL-A-8625C, TYPE 2, CLASS 1

PARTS LIST

TOLERANCES

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RETA N R, MOTOR

D 264 1 2529004

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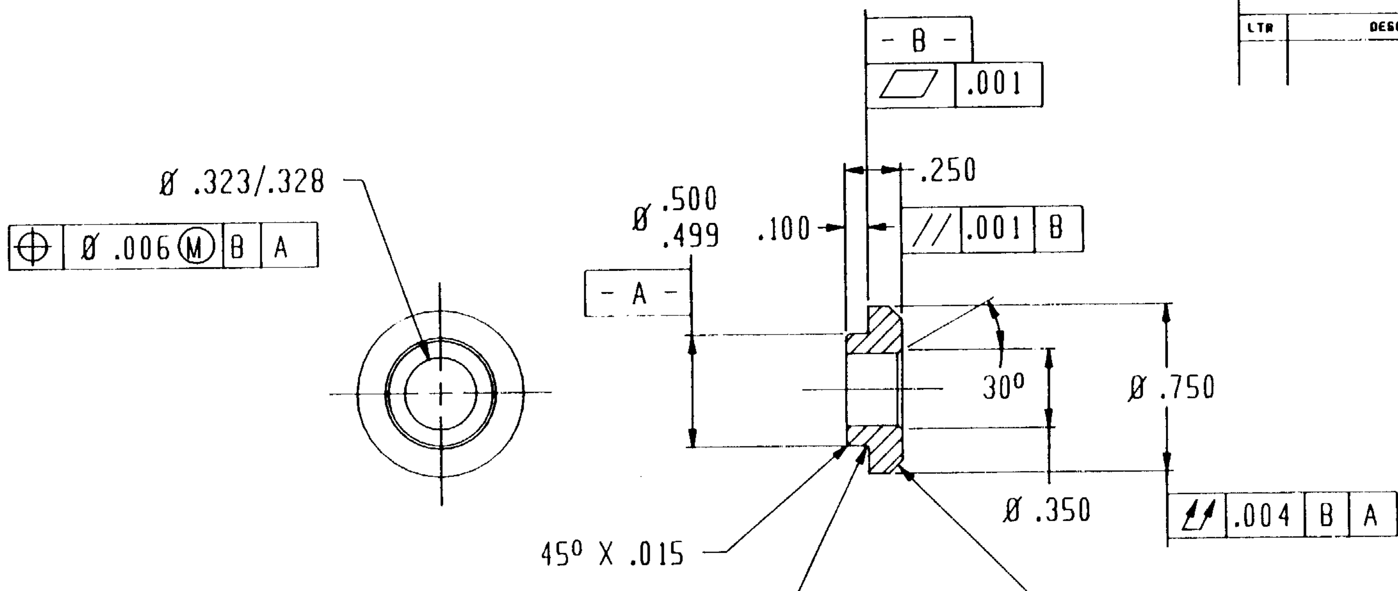

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED

NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND EDGES EQUIV TO R .005
3. SURFACE FINISH 63RMS UNLESS SHOWN

			-1		SST, TYPE 304
UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	(STOCK,SIZE,MATERIAL,SPECIFICATION,TYPE,CLASS,GRADE,HEAT TREAT) NOT TEXT

PARTS LIST

NEXT ASSY.	USED ON

APPLICATION

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UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES AND INCLUDE CHEMICALLY APPLIED OR PLATED FINISHES

TOLERANCES

BASIC DIMENSION	DECIMALS 2 PLACE	DECIMALS 3 PLACE	FRACTIONS
UNDER 6	±.01	±.005	±1/32
6-24 INCL.	±.03	±.010	±1/16
OVER 24	±.06	±.015	±1/8
ANGLES	± 0°-30'		

COML TOL APPLY TO STOCK SIZES

CONTRACT		
DRN	E. C. SCHROEDER	2-8-90
CHK	E. C. SCHROEDER	9-21-90
MECH.		
ELEC.		
PROJ.		

SOUTHWEST RESEARCH INSTITUTE

WASHER, MOTOR

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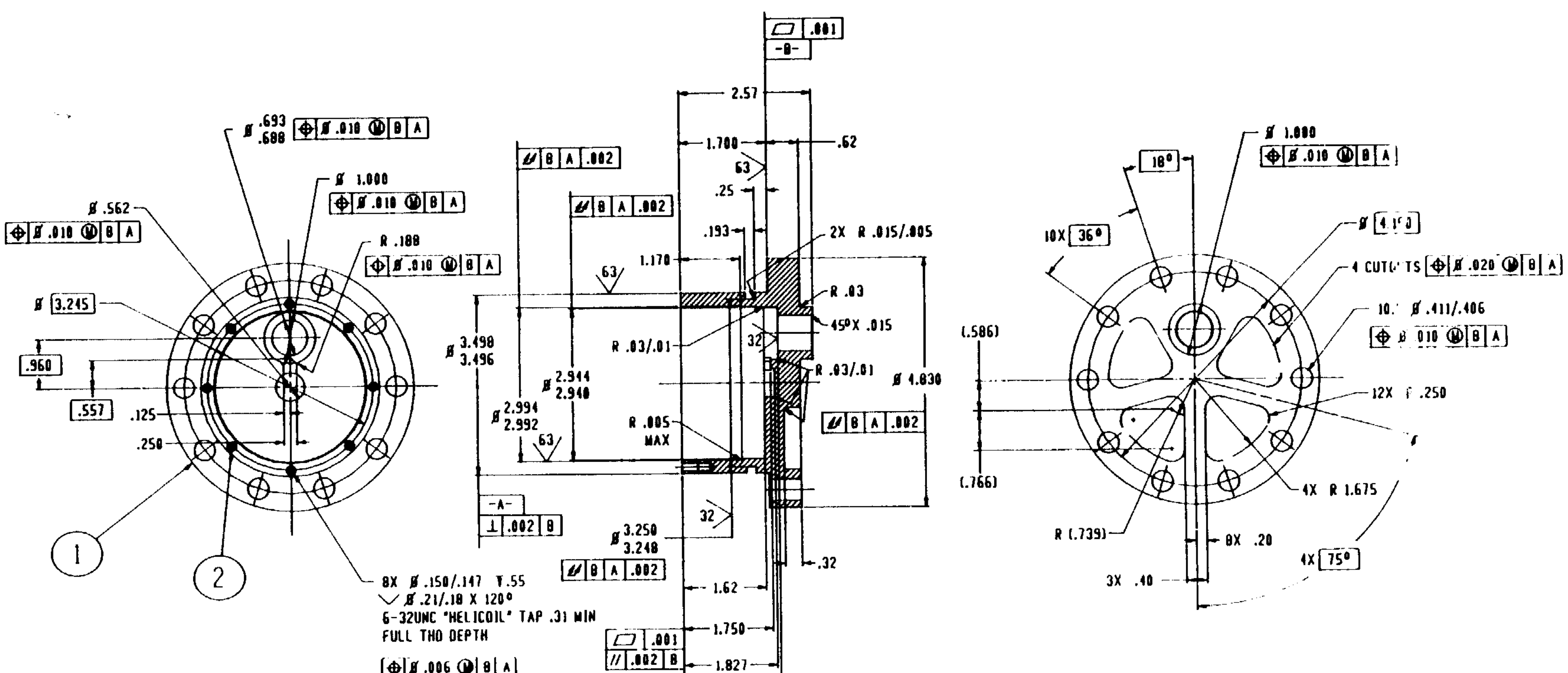

NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982

2. RMV ALL BURRS AND EDGES EQUIV TO R.005

3. SURFACE FINISH 125RMS UNLESS SHOWN

4. ANODIZE PER MIL-A-8625C, TYPE 2, CLASS I

8	EA	2		3585-06CNW275	THD INSERT	HELI-COIL DIVISION
1	EA	1		-2	FLANGE	ALUMINUM, 6061-T6

PARTS LIST

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SOUTHWEST RESEARCH INSTITUTE
SAN ANTONIO, TEXAS

FLANGE ASSY, REAR

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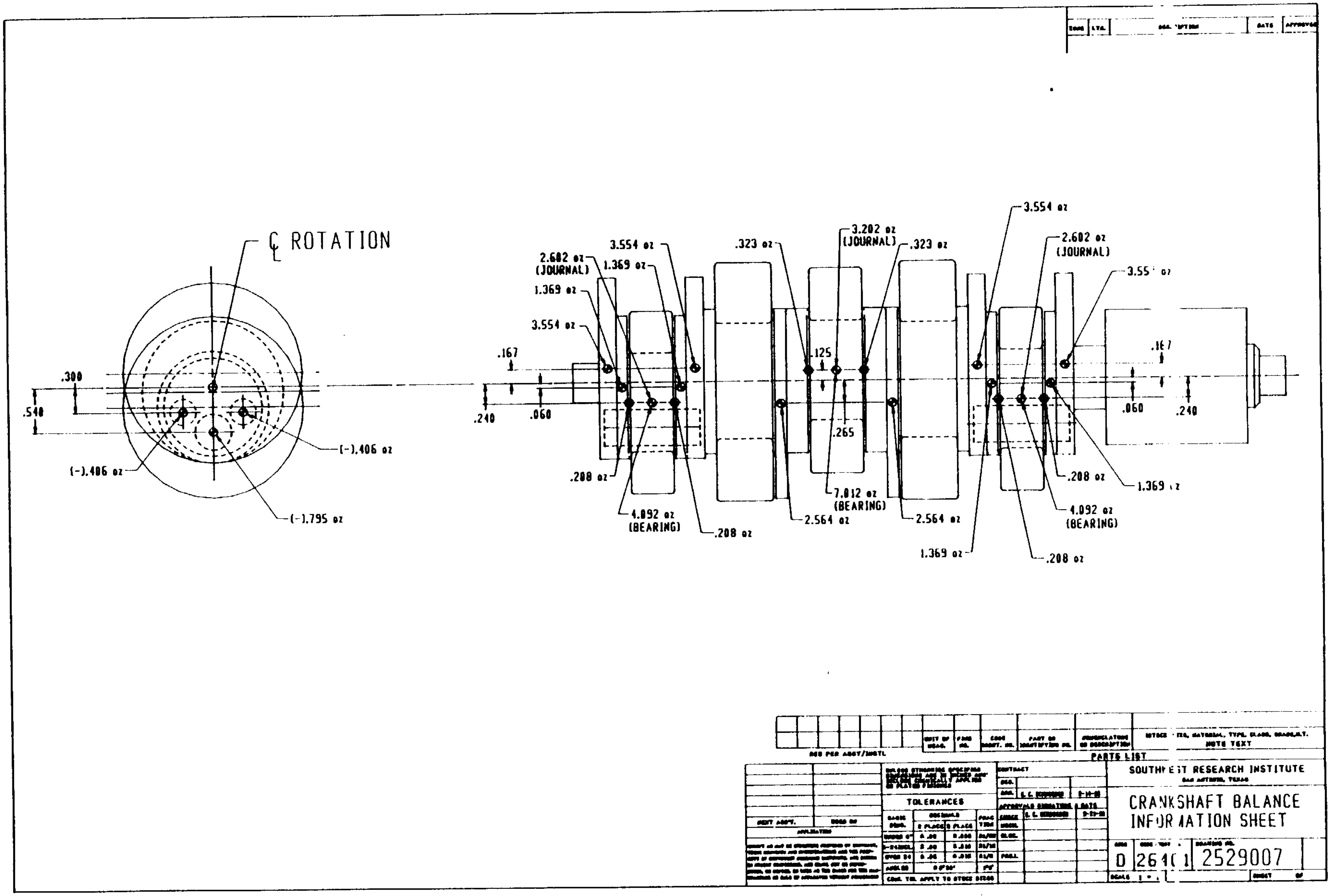

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.540
(-).406 oz
(-).406 oz
(-).795 oz
.167
.240
.060
2.602 oz (JOURNAL)
1.369 oz
3.554 oz
1.369 oz
3.554 oz
.208 oz
4.092 oz (BEARING)
.208 oz
.323 oz
3.202 oz (JOURNAL)
.323 oz
.125
.265
7.012 oz (BEARING)
2.564 oz
2.564 oz
3.554 oz
2.602 oz (JOURNAL)
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.240
.208 oz
1.369 oz
4.092 oz (BEARING)
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SOUTHWEST RESEARCH INSTITUTE
CRANKSHAFT BALANCE
INFORMATION SHEET
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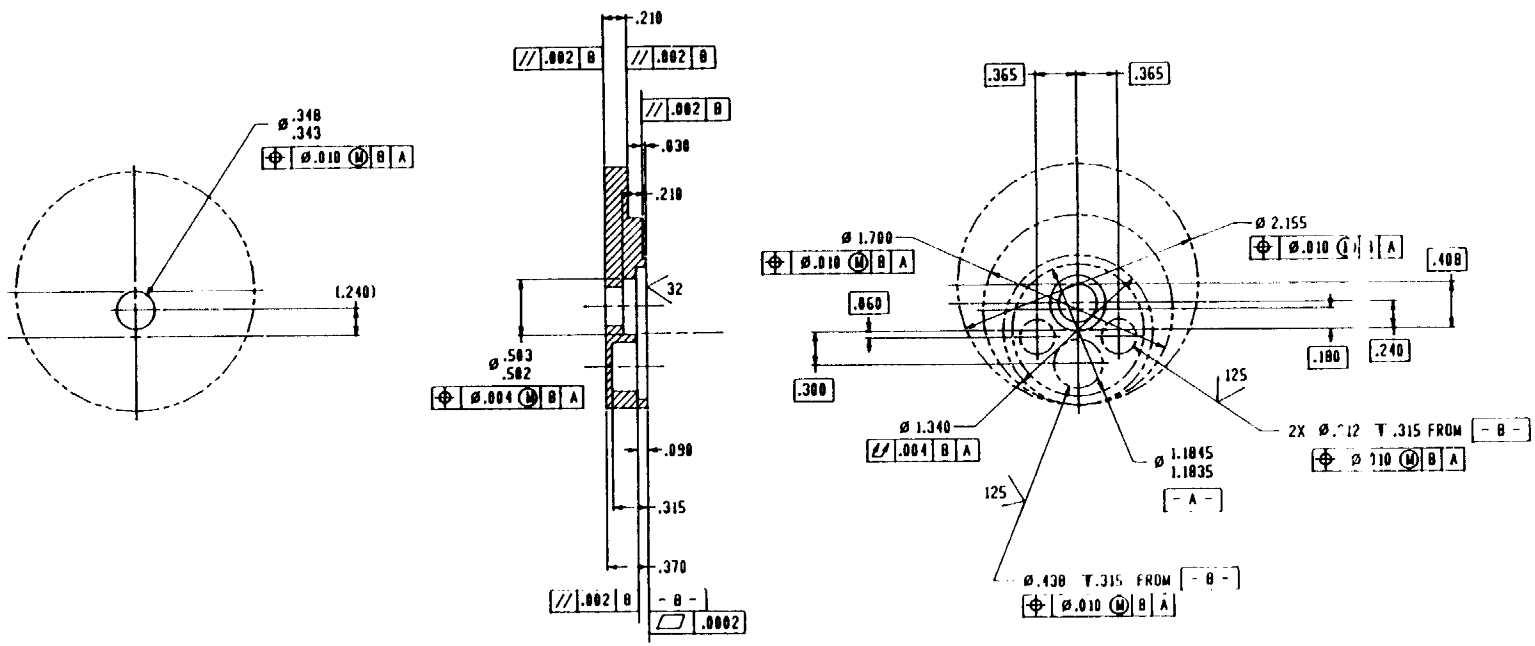

NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND EDGES EQUIV TO R.005
3. SURFACE FINISH 63RMS UNLESS SHOWN
4. PASSIVATE PER MIL-STD-171C, FINISH 5.4.1

-1	SST, TYPE 304		
PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION		
PARTS LIST			
SOUTHWEST RESEARCH INSTITUTE			
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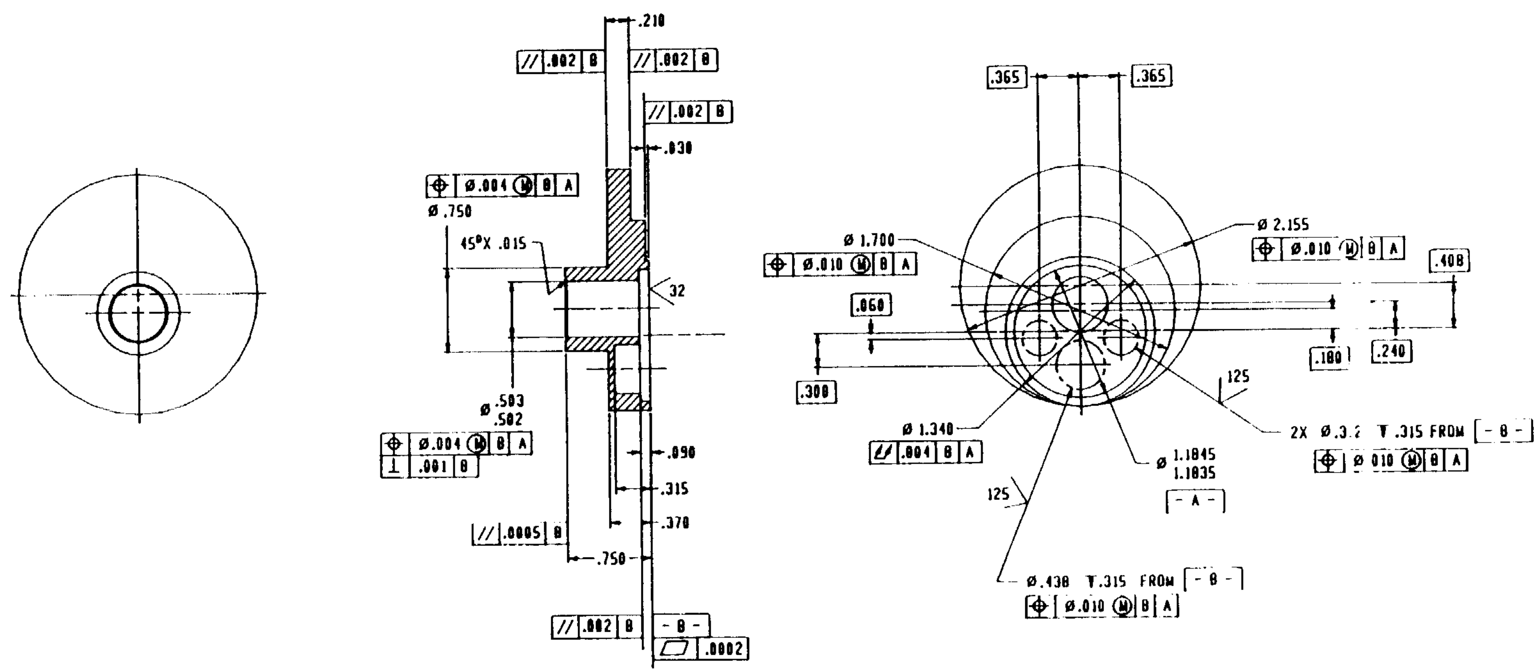
TOLERANCES



NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND EDGES EQUIV TO R.005
3. SURFACE FINISH 63RMS UNLESS SHOWN
4. PASSIVATE PER MIL-STD-171C, FINISH 5.4.1

-1

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TOLERANCES

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RETAINER, REAR BEARING

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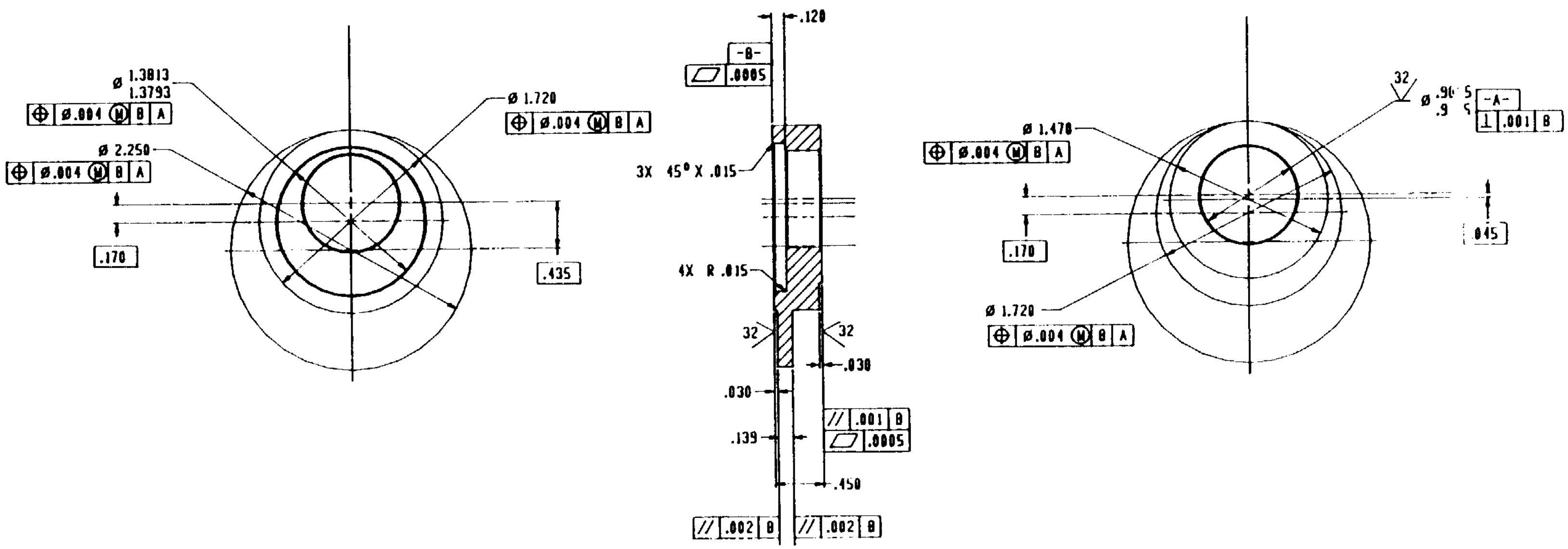

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1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND SHARP EDGES EQUIV TO R.005
3. SURFACE FINISH 63RMS UNLESS SHOWN
4. PASSIVATE PER MIL-STD-171C, FINISH 5.4.1

SST TYPE 304

PARTS LIST

TOLERANCES

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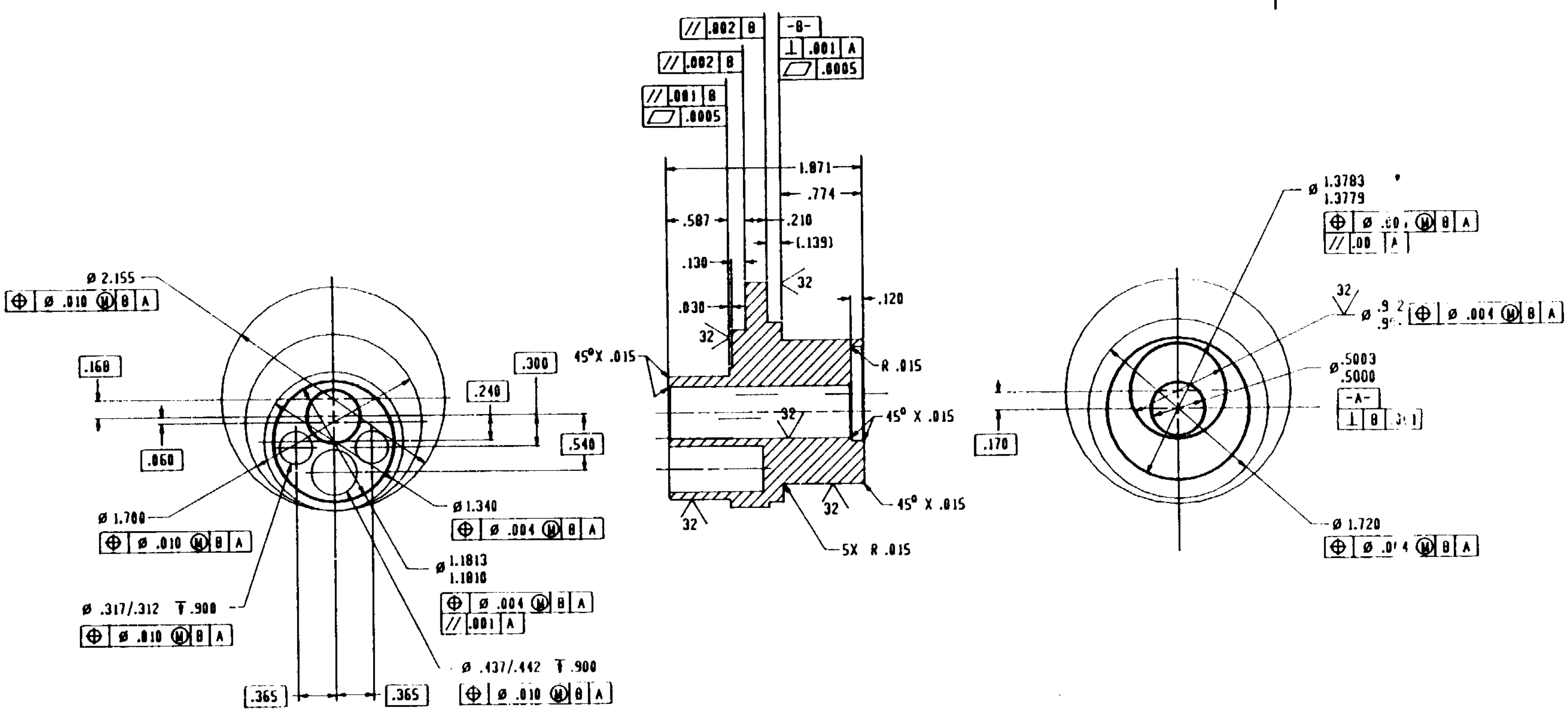
SPACER CRANKSHAFT

D 26411 2529010



NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND EDGES EQUIV TO R.005
3. SURFACE FINISH 63RMS UNLESS SHOWN
4. PASSIVATE PER MIL-STD-171C, FINISH 5.4.1

-1	SST TYPE 304
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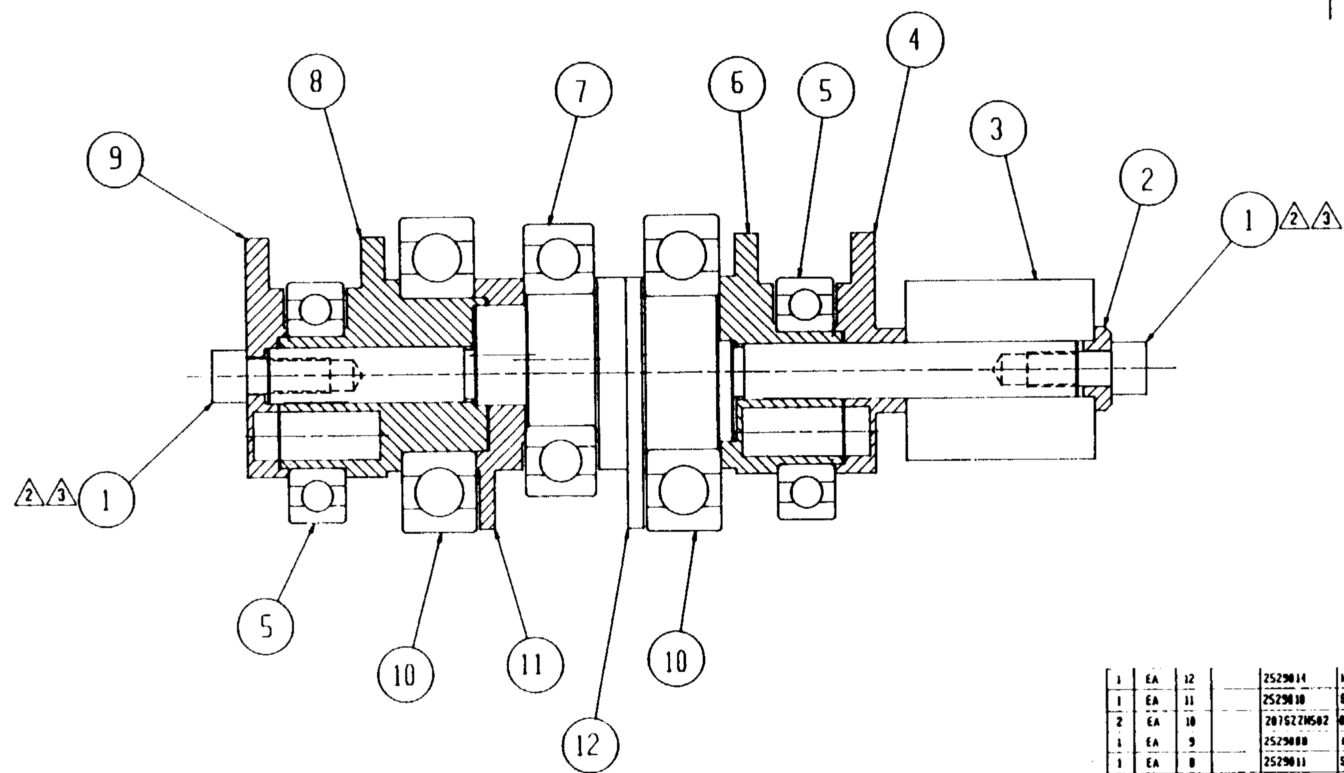


NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. APPLY A THIN COAT OF ANTI-SEIZE COMPOUND TO THE THREADS AND UNDER THE HEADS OF THE SCREWS BEFORE TIGHTENING
3. TIGHTEN SCREWS TO 142 LB-IN TORQUE

QTY	UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	MATERIAL
1	EA	12		2529014	MAIN SHAFT	
1	EA	11		2529010	SPACER	
2	EA	10		207SZZMS02	BEARING	MRC
1	EA	9		2529008	RETAINER	
1	EA	8		2529011	SUPPORT	
1	EA	7		206SZZMS02	BEARING	MR
1	EA	6		2529013	SUPPORT	
2	EA	5		106SZZMS02	BEARING	MRC
1	EA	4		2529009	RETAINER	
1	EA	3		80E01004	ROTOR	INLAND MOTOR
1	EA	2		2529005	WASHER	
2	EA	1			SCREW	.3125-24UNF-2A SOCKET HEAD CAP, 304 SST

PARTS LIST

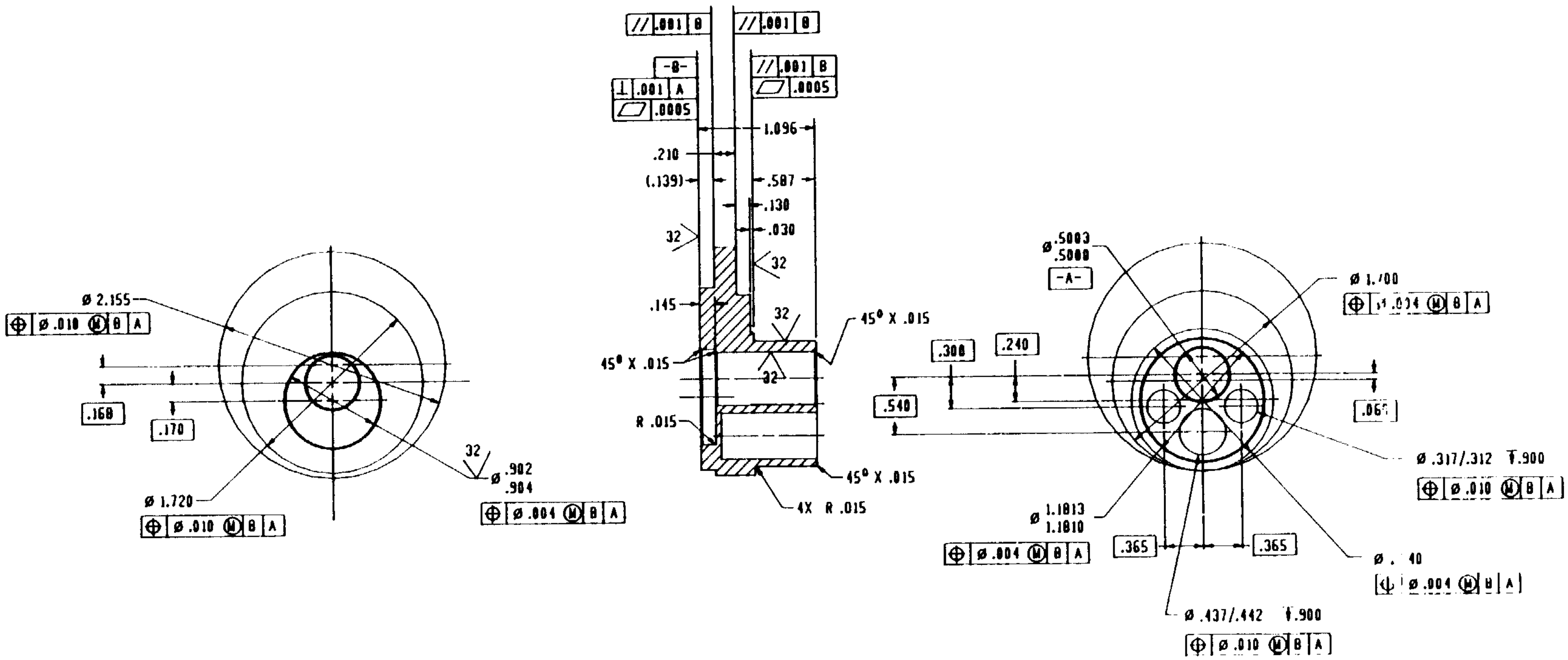
SOUTHWEST RESEARCH INSTITUTE
SAN ANTONIO, TEXAS

CRANKSHAFT ASSEMBLY

TOLERANCES

D | 26401 | 2529012

SCALE 2 : 1



NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND EDGES EQUIV TO R.005
3. SURFACE FINISH 63RMS UNLESS SHOWN
4. PASSIVATE PER MIL-STD-171C, FINISH 5.4.1

-1 SST TYPE 304

PARTS LIST

TOLERANCES

SOUTHWEST RESEARCH INSTITUTE
SAN ANTONIO, TEXAS

BEARING SUPPORT, REAR

D 26411 2529013

SCALE 2 - 1

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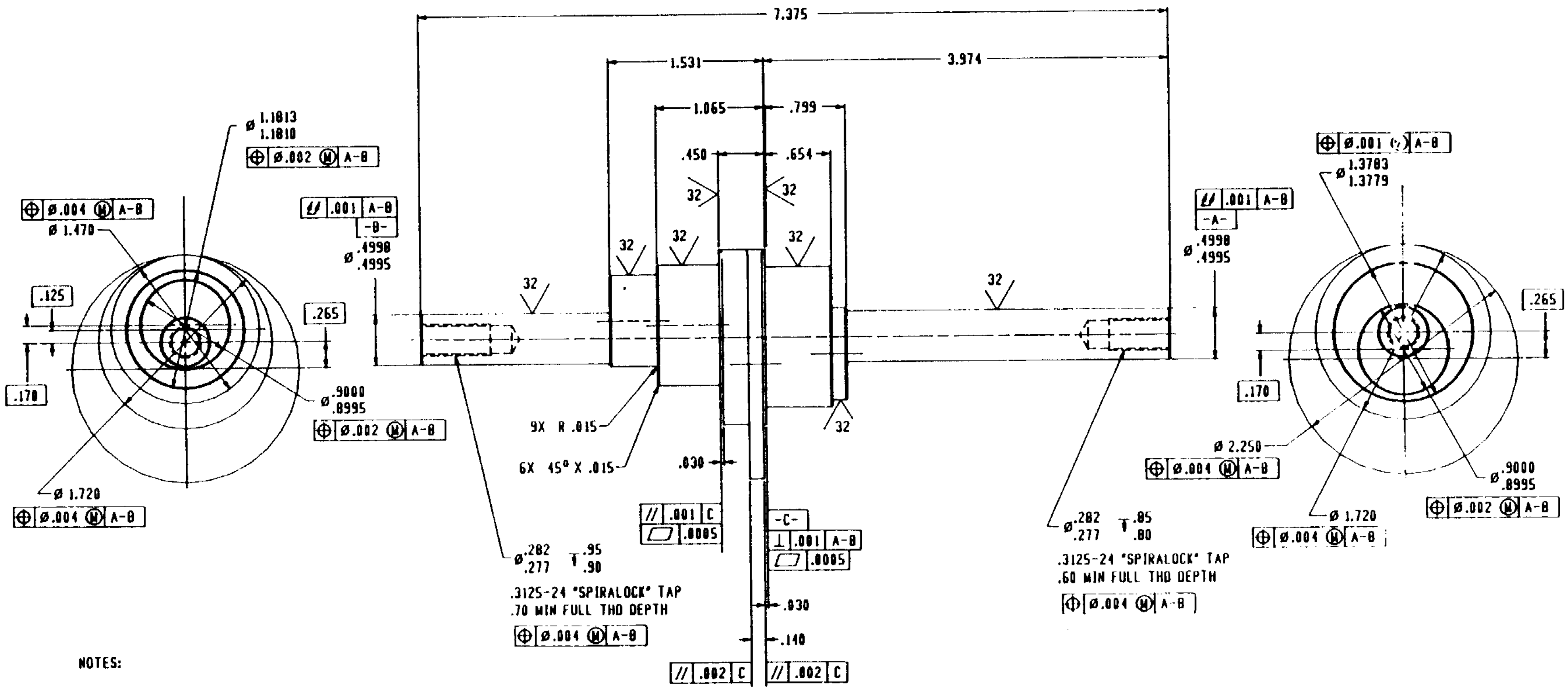

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1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND SHARP EDGES EQUIV TO R.005
3. SURFACE FINISH 63RMS UNLESS SHOWN
4. PASSIVATE PER MIL-STD-171C, FINISH 5.4.1

-1 | SS TYPE 304

PARTS LIST

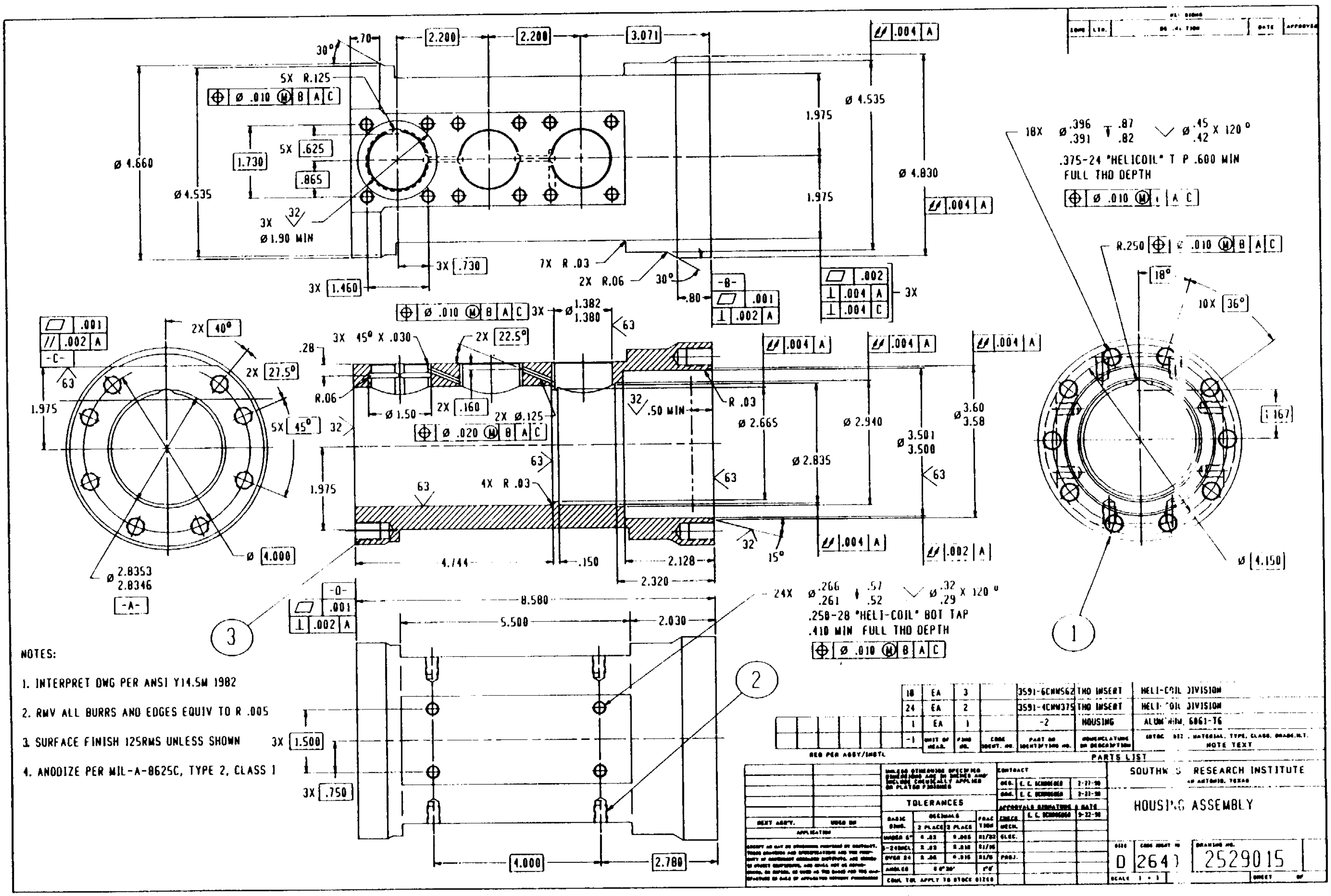
TOLERANCES

SOUTHWEST RESEARCH INSTITUTE

MAIN SHAFT

D 2529014

SCALE 2



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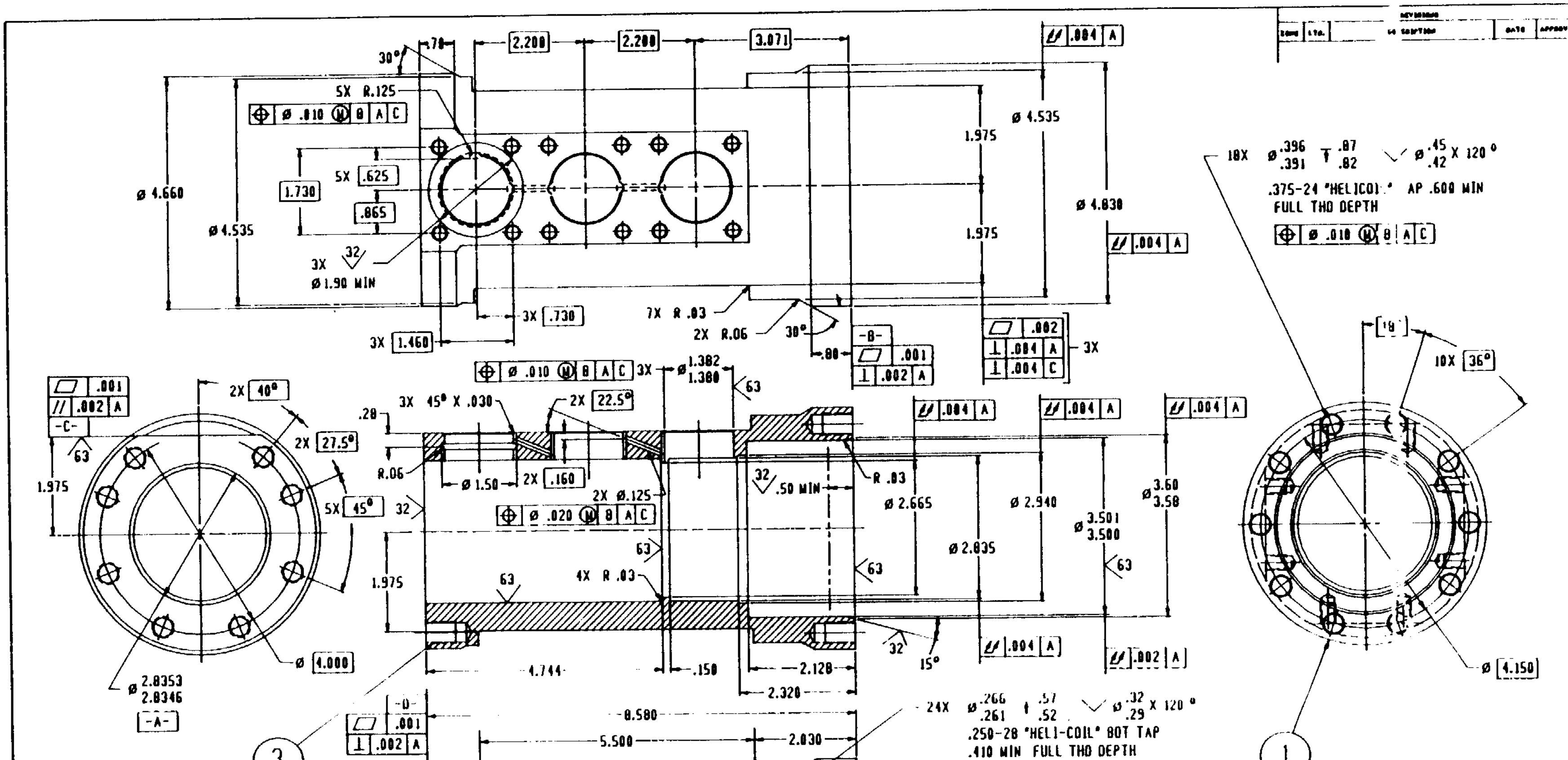


NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND EDGES EQUIV TO R .005
3. SURFACE FINISH 125RMS UNLESS SHOWN
4. ANODIZE PER MIL-A-8625C, TYPE 2, CLASS I

18	EA	3		3591-6CNW562	THD INSERT	HELI-COIL DIVISION
24	EA	2		3591-4CNW375	THD INSERT	HELI-COIL DIVISION
1	EA	1		-2	HOUSING	ALUMINUM 6061-T6

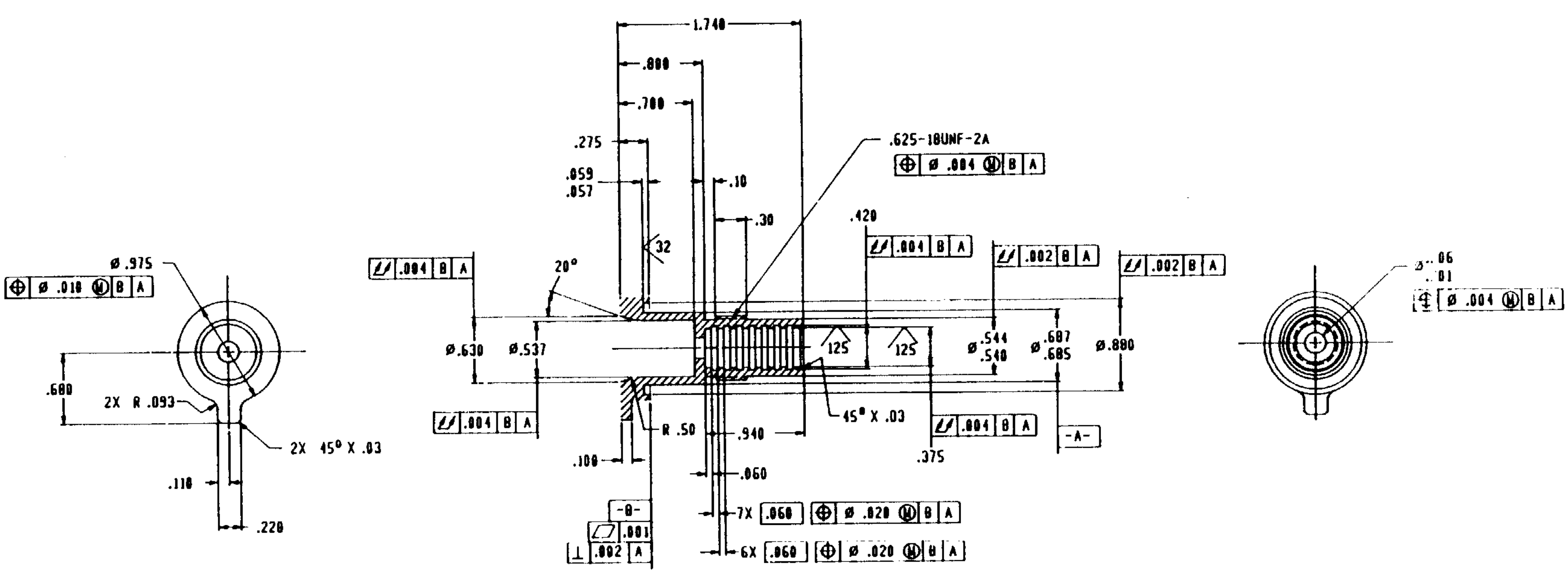
PARTS LIST

TOLERANCES

SOUTHWEST RESEARCH INSTITUTE

HOUSING ASSEMBLY

D 26401 2529015



NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND EDGES EQUIV TO R .005
3. SURFACE FINISH 63 RMS UNLESS SHOWN
4. ANODIZE PER MIL-A-8625C, TYPE 2, CLASS 1

-1	ALUMINUM, 6061-T6
PARTS LIST	

TOLERANCES

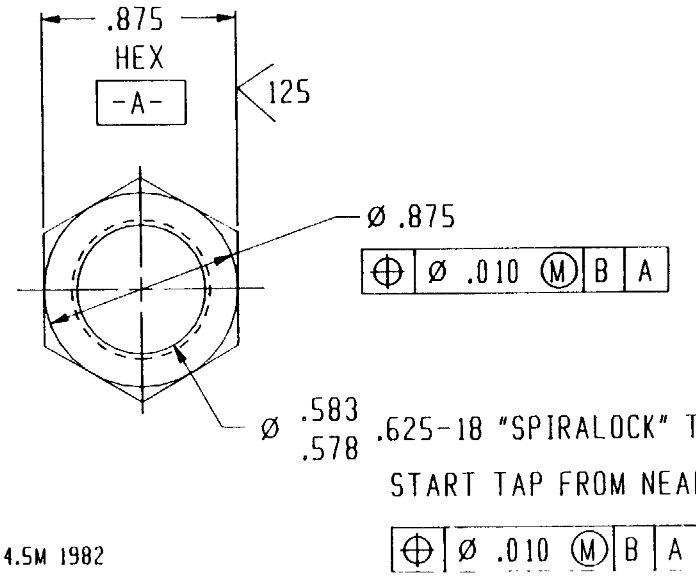
SOUTHWEST RESEARCH INSTITUTE

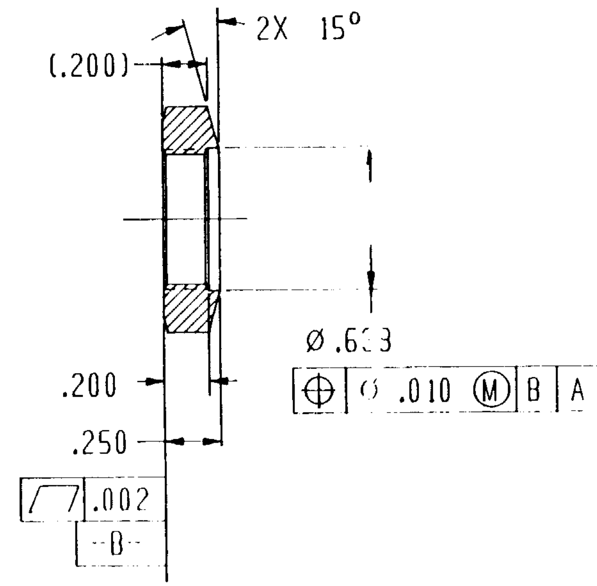
FEEDTHROUGH, ELECTRICAL

D 26401 2529016

SCALE 2:1

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED





NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND EDGES EQUIV TO R .005
3. SURFACE FINISH 63 RMS UNLESS SHOWN
4. ANODIZE PER MIL-A-8625C, TYPE 2, CLASS I

UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	(STOCK, SIZE, MATERIAL, SPECIFICATION, TYPE, CLASS, GRADE, HEAT TREAT) NOTE TEXT
			-1		ALUMINUM, 6061-T6

PARTS LIST

NEXT ASSY.	USED ON

APPLICATION

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UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES AND INCLUDE CHEMICALLY APPLIED OR PLATED FINISHES

TOLERANCES

BASIC DIMENSION	DECIMALS 2 PLACE	DECIMALS 3 PLACE	FRACTIONS
UNDER 6	± .01	± .005	± 1/32
6-24 INCL.	± .03	± .010	± 1/16
OVER 24	± .06	± .015	± 1/8
ANGLES	± 0°-30'		

COML TOL APPLY TO STOCK SIZES

CONTRACT		
DRN	E. C. SCHROEDER	3-7-90
CHK	E. C. SCHROEDER	9-22-90
MECH.		
ELEC.		
PROJ.		

SOUTHWEST RESEARCH INSTITUTE

NUT, FEEDTHROUGH

SIZE	CODE IDENT. NO.	DRAWING NO.
B	26401	2529017

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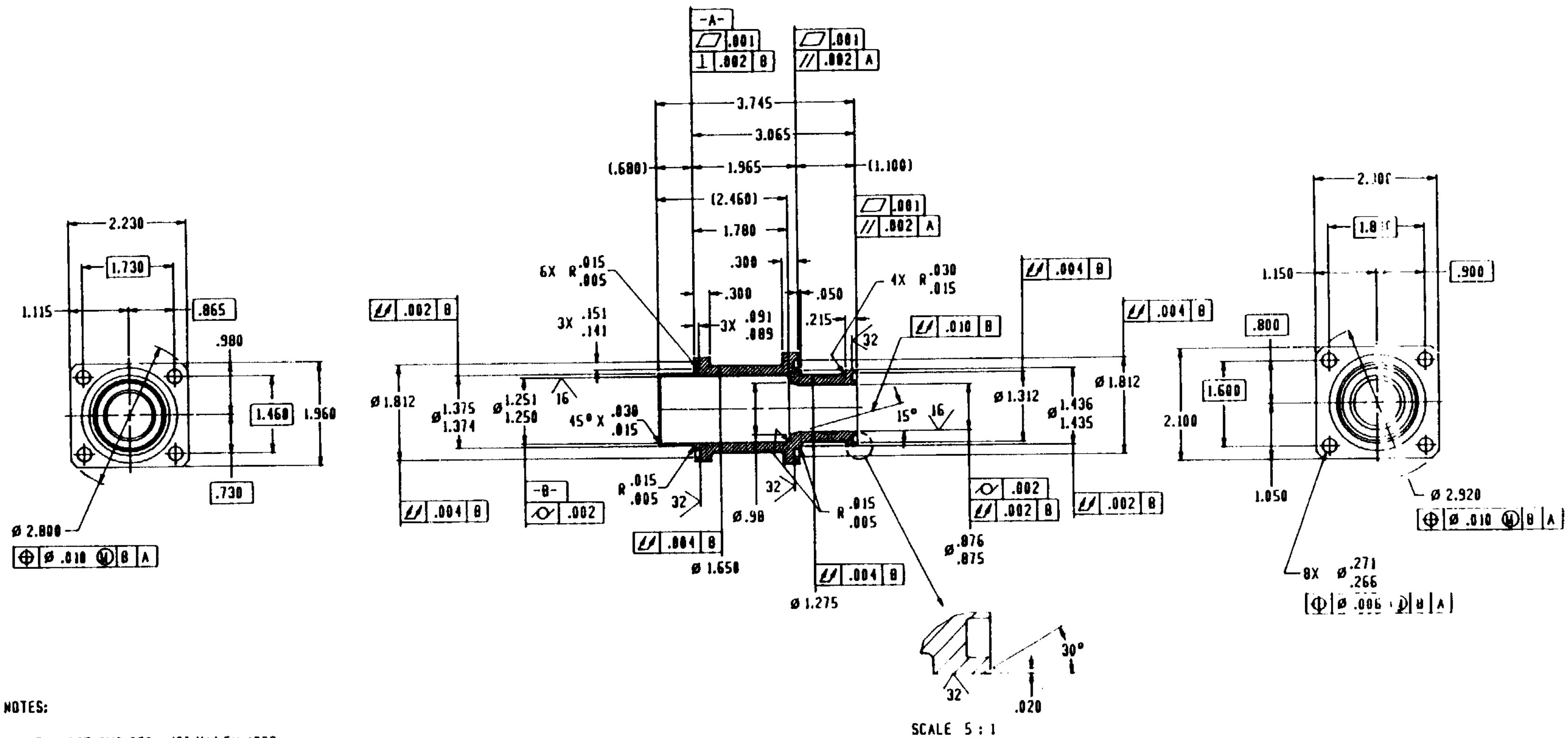


NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND SHARP EDGES EQUIV TO R.005
3. SURFACE FINISH 63RMS UNLESS SHOWN
4. TREAT SURFACE OF PART PER MIL-A-6357A (AR)
   APPROVED SOURCE: TIODIZE CO., INC.
   HUNTINGTON BEACH, CA
   "HARDTUF X20" PROCESS
5. SURFACE TREATMENT WILL ADD .001 ±.0002 TO EACH SURFACE

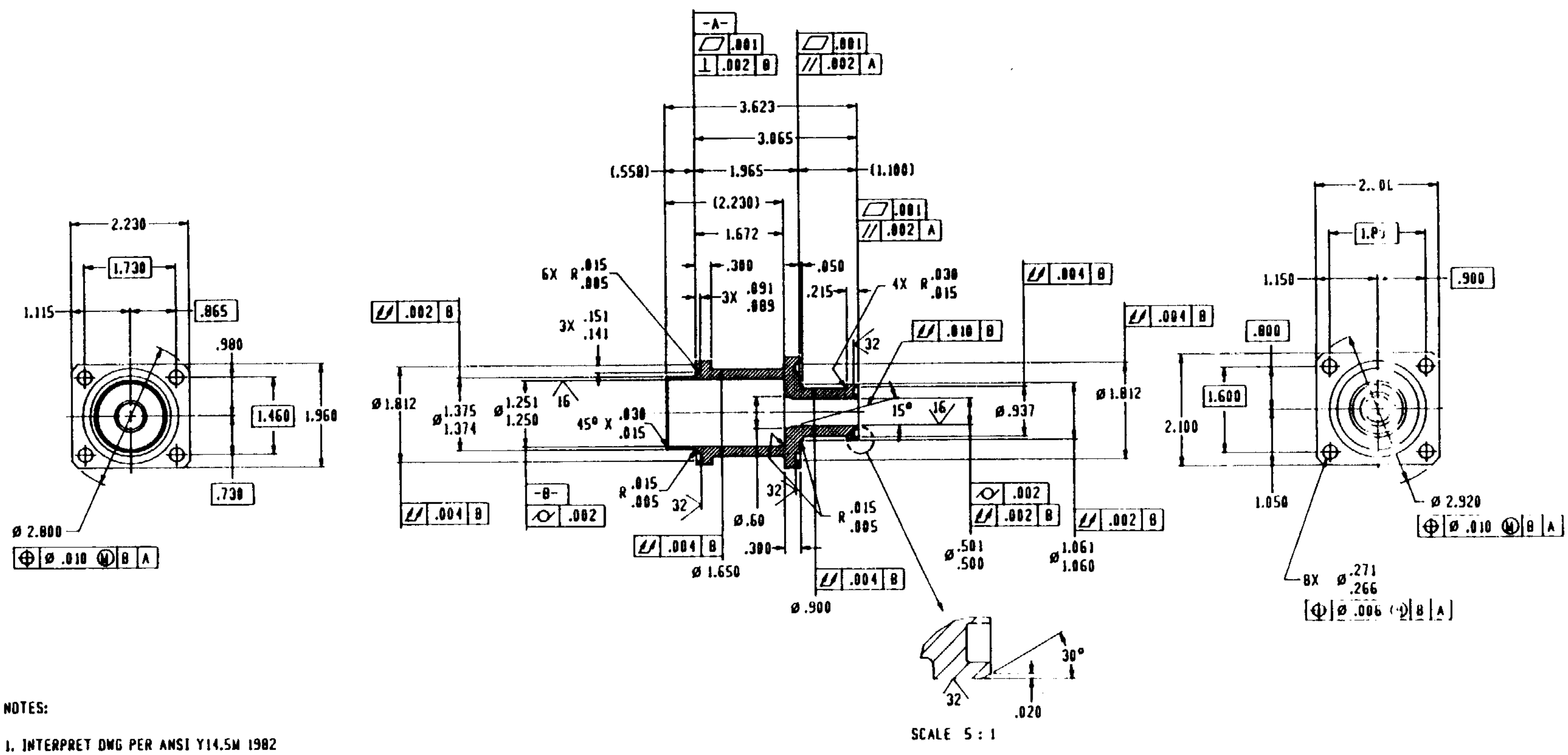
ALUMINU, 6061-T6

PARTS LIST

SOUTHWEST RESEARCH INSTITUTE

CYLINDER FIRST STAGE

D 26101 2529018



NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982

2. RMV ALL BURRS AND SHARP EDGES EQUIV TO R.005

3. SURFACE FINISH 63RMS UNLESS SHOWN

4. TREAT SURFACE OF PART PER MIL-A-6357A (AR)
   APPROVED SOURCE: TIODIZE CO., INC.
   HUNTINGTON BEACH, CA
   "HARDTUF X20" PROCESS

5. SURFACE TREATMENT WILL ADD .001 ±.0002 TO EACH SURFACE

ALUMIN. AL 6061-T6

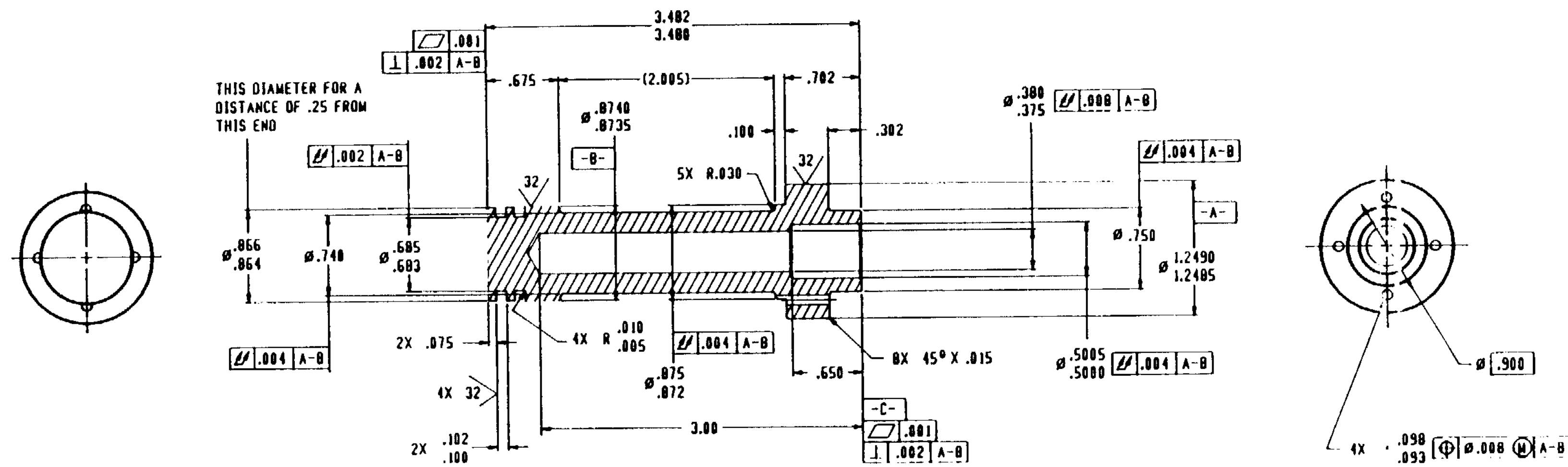
PARTS LIST

TOLERANCES

SOUTHWEST RESEARCH INSTITUTE

CYLINDER, SECOND STAGE

D 264.1 2529019



NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND EDGES EQUIV TO R.005
3. SURFACE FINISH 63RMS UNLESS SHOWN

TORLON 4301, AMOCO CHEM CORP

PARTS LIST

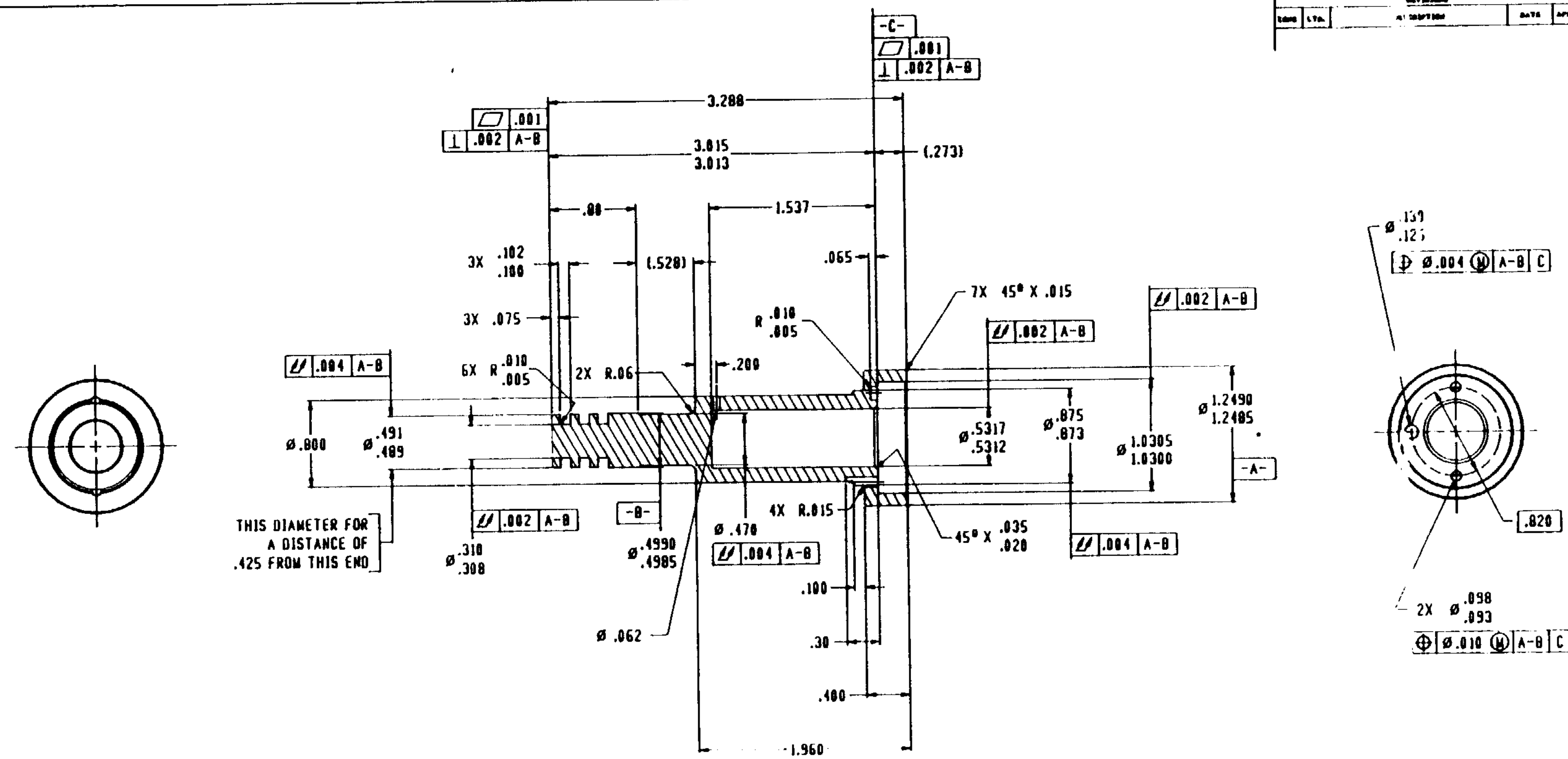
TOLERANCES

SOUTHWEST RESEARCH INSTITUTE

PISTON, FIRST STAGE

D 26401 2529020

SCALE 2:1



NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND EDGES EQUIV TO R.005
3. SURFACE FINISH 63RMS UNLESS SHOWN

TORLON 301, AMOCO CHEM CORP

PARTS LIST

TOLERANCES

SOUTHWEST RESEARCH INSTITUTE

PISTON, SECOND STAGE

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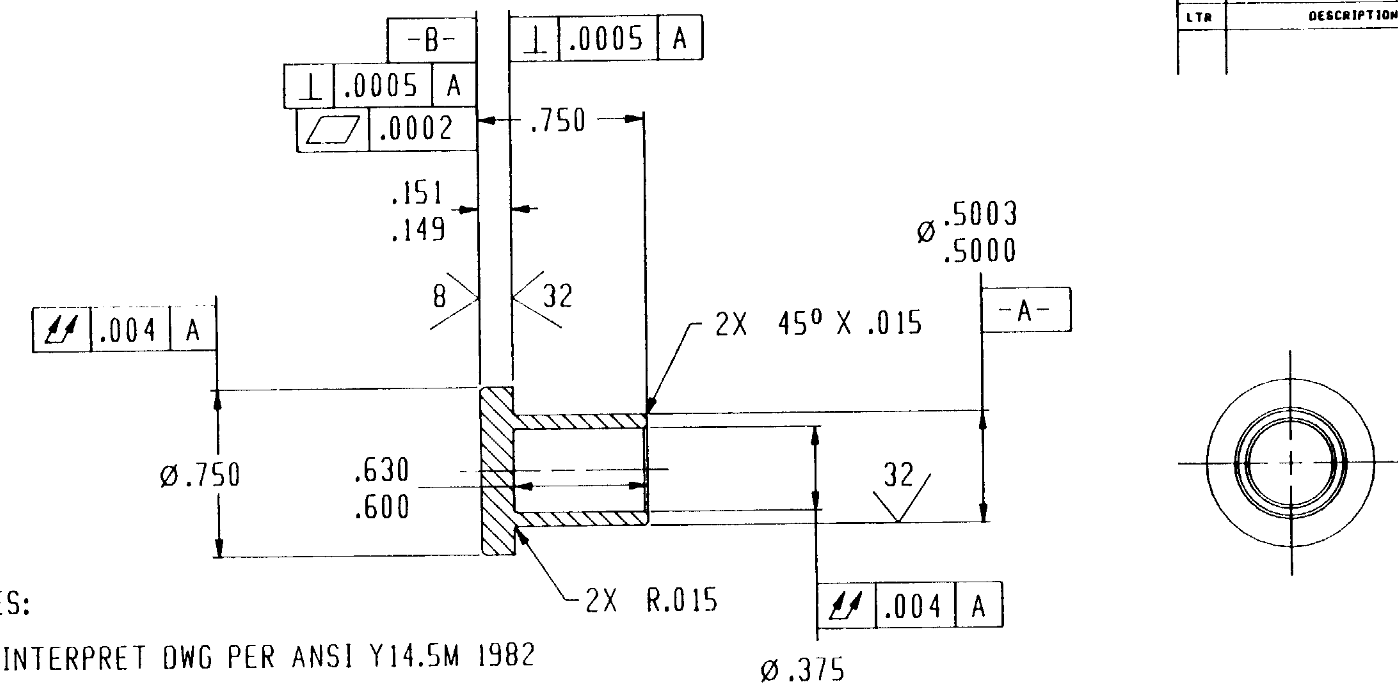

NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND EDGES EQUIV TO R.005
3. SURFACE FINISH 63RMS UNLESS SHOWN
4. HARDENING TEMP 1850-1950 °F. AIR COOL. DRAW AT 800°F FOR 1 HOUR. RC 55-58.

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED

PARTS LIST

UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	NOTE TEXT (STOCK,SIZE,MATERIAL,SPECIFICATION,TYPE,CLASS,GRADE,HEAT TREAT)
			-1		4. SST, TYPE 440-C

NEXT ASSY.	USED ON
APPLICATION	

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TOLERANCES

BASIC DIMENSION	DECIMALS 2 PLACE	DECIMALS 3 PLACE	FRACTIONS
UNDER 6	±.01	±.005	±1/32
6-24 INCL.	±.03	±.010	±1/16
OVER 24	±.06	±.015	±1/8
ANGLES	± 0°-30'		

COML TOL APPLY TO STOCK SIZES

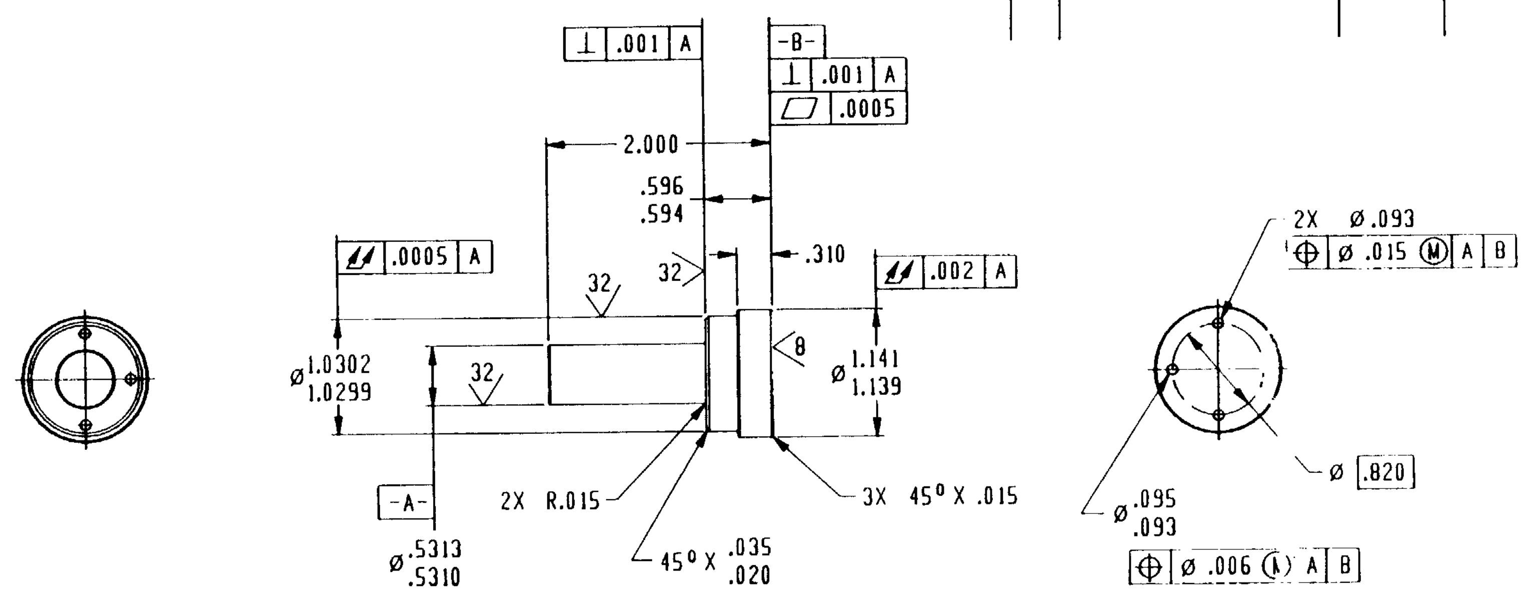
CONTRACT		
DRN	E. C. SCHROEDER	4-19-90
CHK	E. C. SCHROEDER	9-23-90
MECH.		
ELEC.		
PROJ.		

SOUTHWEST RESEARCH INSTITUTE

FOLLOWER, FIRST STAGE PISTON

SIZE	CODE IDENT. NO.	DRAWING NO.
B	26401	2529022
SCALE 2 : 1		SHEET OF

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED



NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND EDGES EQUIV TO R.005
3. SURFACE FINISH 63RMS UNLESS SHOWN

UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	(STOCK,SIZE,MATERIAL,SPECIFICATION,TYPE,CLASS,GRADE,HEAT TREAT) NOTE TEXT
			-1		TUNGSTEN ALLOY, 97%W, 2.1%Ni, .9%Fe

PARTS LIST

NEXT ASSY.	USED ON

APPLICATION

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TOLERANCES

BASIC DIMENSION	DECIMALS 2 PLACE	DECIMALS 3 PLACE	FRACTIONS
UNDER 6	±.01	±.005	±1/32
6-24 INCL.	±.03	±.010	±1/16
OVER 24	±.06	±.015	±1/8
ANGLES	± 0°-30'		

COML TOL APPLY TO STOCK SIZES

CONTRACT		
DRN	E. C. SCHROEDER	4-20-90
CHK	E. C. SCHROEDER	9-23-90
MECH.		
ELEC.		
PROJ.		

SOUTHWEST RESEARCH INSTITUTE

FOLLOWER, SECOND STAGE PISTON

SIZE	CODE IDENT. NO.	DRAWING NO.
B	26401	2529023
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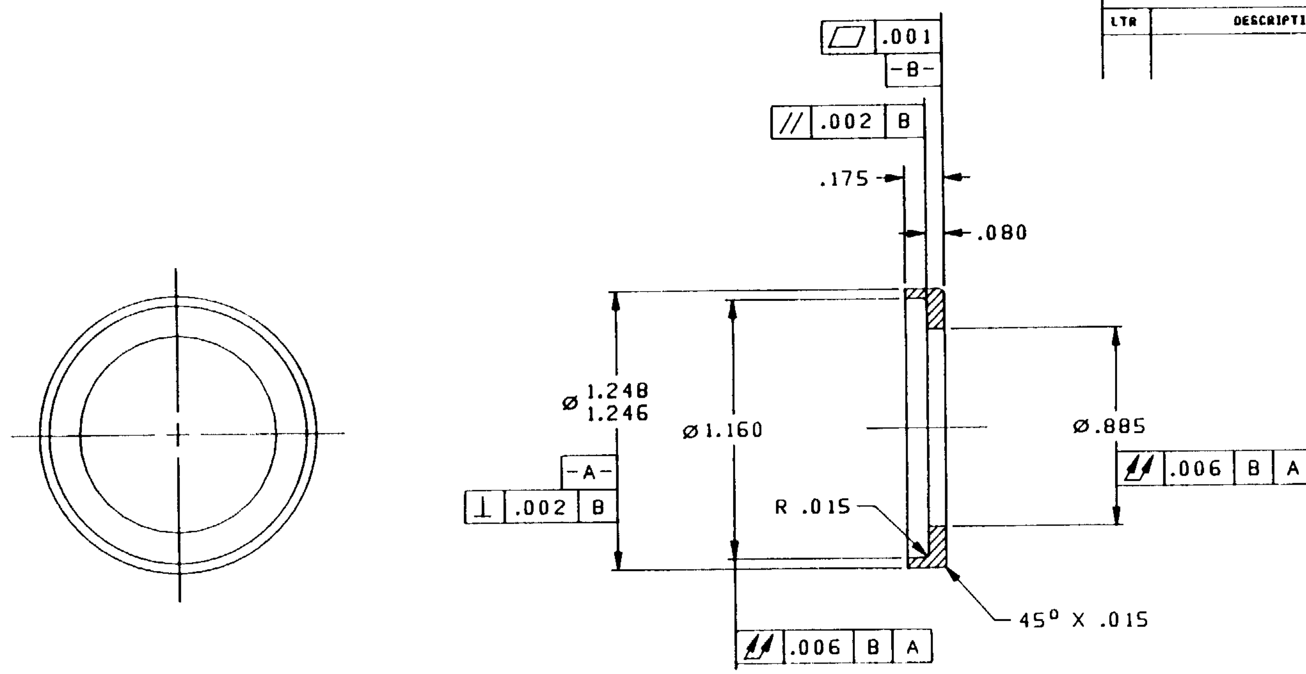

NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND SHARP EDGES EQUIV TO R .005
3. SURFACE FINISH 63RMS
4. ANODIZE PER MIL-STD-8625C, TYPE 2, CLASS I

LTR	DESCRIPTION	DATE	APPROVED

REVIS[illegible]NS

UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	(STOCK,SIZE,MATERIAL,SPECIFICA[illegible]N,TYPE,CLASS,GRADE,HEAT TREAT) NOT[illegible] TEXT
			-1		AL, 6061-T6

PARTS LIST

NEXT ASSY.	USED ON

APPLICATION

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UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES AND INCLUDE CHEMICALLY APPLIED OR PLATED FINISHES

TOLERANCES

BASIC DIMENSION	DECIMALS 2 PLACE	DECIMALS 3 PLACE	FRACTIONS
UNDER 6	±.01	±.005	±1/32
6-24 INCL.	±.03	±.010	±1/16
OVER 24	±.06	±.015	±1/8
ANGLES	± 0°-30'		

COML TOL APPLY TO STOCK SIZES

CONTRACT		
DRN	E. C. SCHROEDER	4-24-90
CHK	E. C. SCHROEDER	9-23-90
MECH.		
ELEC.		
PROJ.		

SOUTHWEST RE[illegible]EARCH INSTITUTE

SEAT, PISTO[illegible] SPRING

SIZE	CODE IDENT. NO.	DR[illegible]ING NO.
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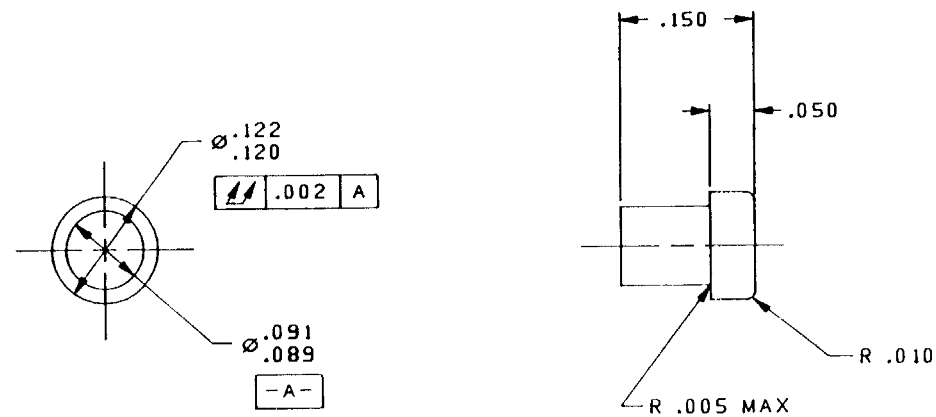
REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED



NOTES:

1. INTERPRET DWG PER ANSI 14.5M 1982
2. RMV ALL BURRS AND SHARP EDGES EQUIV TO R .005
3. SURFACE FINISH 63RMS
4. PASSIVATE PER MIL-STD-171C, FINISH 5.4.1

UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	(STOCK, SIZE, MATERIAL, SPECIFICATION, TYPE, CLASS, GRADE, HEAT TREAT) NOT TEXT
			-1		SST, TYPE 304

PARTS LIST

NEXT ASSY.	USED ON

APPLICATION

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TOLERANCES

BASIC DIMENSION	DECIMALS 2 PLACE	DECIMALS 3 PLACE	FRACTIONS
UNDER 6	±.01	±.005	±1/32
6-24 INCL.	±.03	±.010	±1/16
OVER 24	±.06	±.015	±1/8
ANGLES	± 0°-30'		

COML TOL APPLY TO STOCK SIZES

CONTRACT

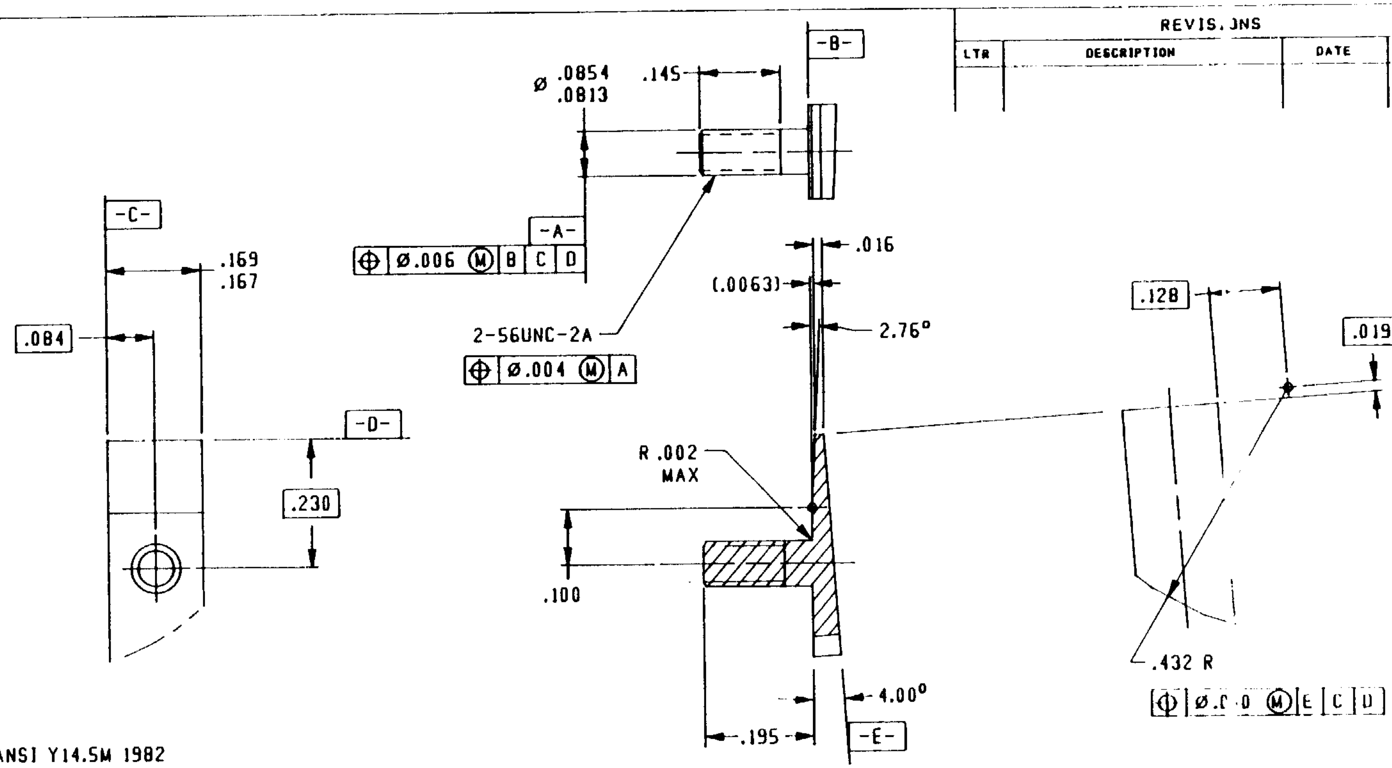
DRN	E. C. SCHROEDER	4-24-90
CHK	E. C. SCHROEDER	9-23-90
MECH.		
ELEC.		
PROJ.		

SOUTHWEST RESEARCH INSTITUTE

PIN, PISTON LOCATING

SIZE	CODE IDENT. NO.	DR. WING NO.
B	26401	2529026

SCALE 8 - 1 SHEET OF



NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV BURRS AND EDGES EQUIV TO R .005
3. SURFACE FINISH 63RMS
4. ANODIZE PER MIL-A-8625C, TYPE 2, CLASS I

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED

UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	(STOCK, SIZE, MATERIAL, SPECIFICATION, TYPE, CLASS, GRADE, HEAT TREAT) NOTE TEXT
			-1		ALUMINUM, 6061-T'

PARTS LIST

NEXT ASSY.	USED ON

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TOLERANCES

BASIC DIMENSION	DECIMALS 2 PLACE	DECIMALS 3 PLACE	FRACTIONS
UNDER 6	±.01	±.005	±1/32
6-24 INCL.	±.03	±.010	±1/16
OVER 24	±.06	±.015	±1/8
ANGLES	± 0°-30'		

COML TOL APPLY TO STOCK SIZES

CONTRACT

DRN	E.C. SCHROEDER	5-3-90
CHK	E.C. SCHROEDER	9-23-90
MECH.		
ELEC.		
PROJ.		

SOUTHWEST RESEARCH INSTITUTE

RETAINER, FIRST STAGE INTAKE VALVE

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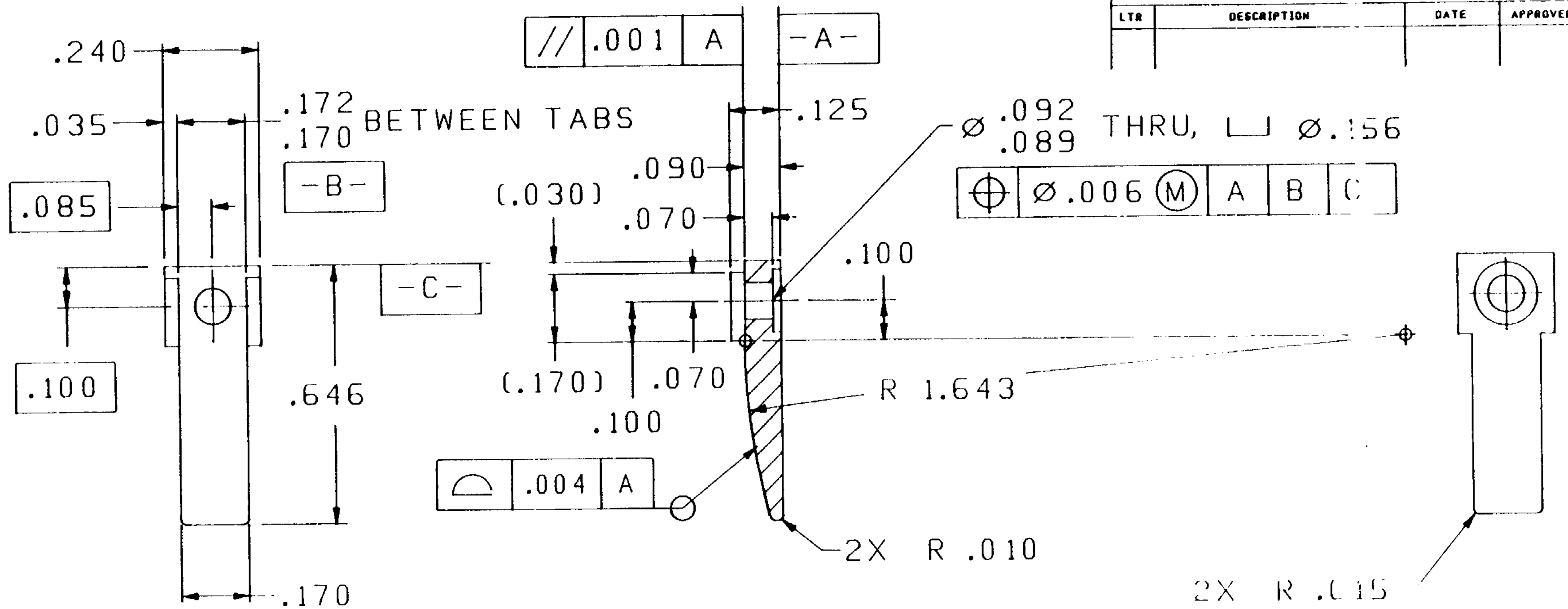

NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV BURRS AND SHARP EDGES EQUIV TO R .005
3. SURFACE FINISH 63 RMS
4. ANODIZE PER MIL-A-8625C, TYPE 2, CLASS 1

UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	(STOCK,SIZE,MATERIAL,SPECIFICATION,TYPE,CLASS,GRADE,HEAT TREAT) NOTE TEXT
			-1		ALUMINUM, 6061-T(

PARTS LIST

NEXT ASSY.	USED ON

APPLICATION

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TOLERANCES

BASIC DIMENSION	DECIMALS 2 PLACE	DECIMALS 3 PLACE	FRACTIONS
UNDER 6	±.01	±.005	±1/32
6-24 INCL.	±.03	±.010	±1/16
OVER 24	±.06	±.015	±1/8
ANGLES	± 0°-30'		

COML TOL APPLY TO STOCK SIZES

CONTRACT		
DRN	E.C. SCHROEDER	5-4-90
CHK	E.C. SCHROEDER	9-23-90
MECH.		
ELEC.		
PROJ.		

SOUTHWEST RESEARCH INSTITUTE

RETAINER, FIRST STAGE OUTLET VALVE

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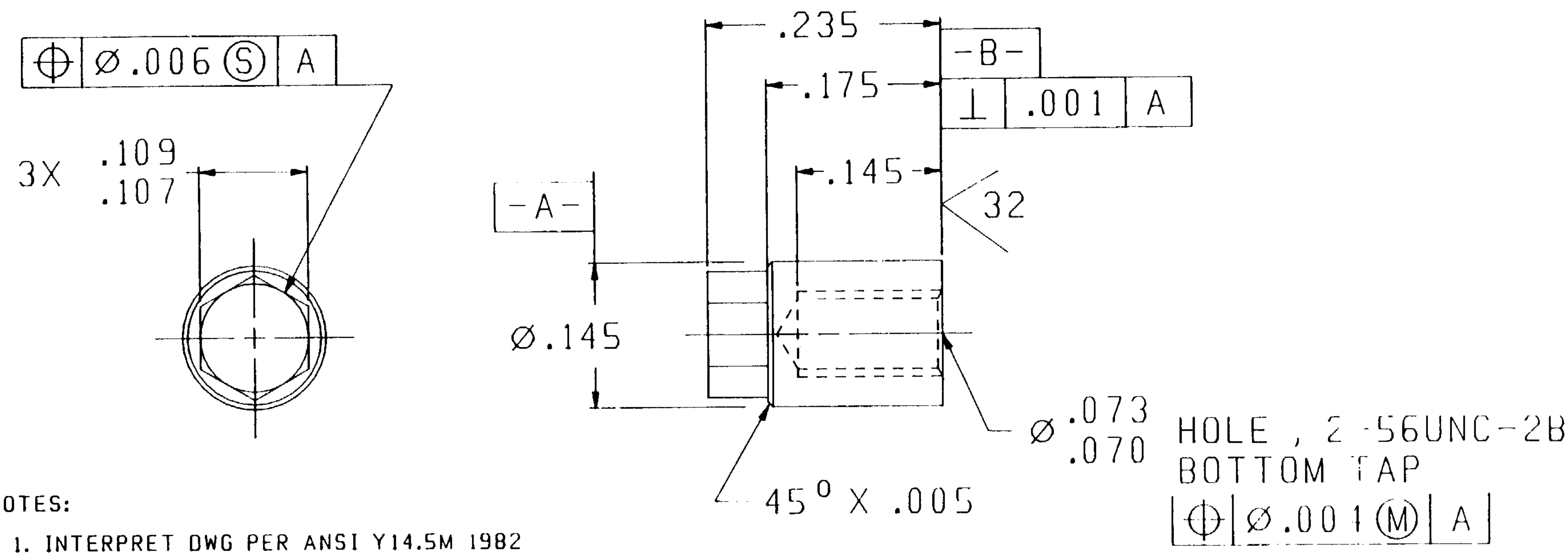

NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV BURRS AND EDGES EQUIV TO R .005
3. SURFACE FINISH 63 RMS UNLESS SHOWN
4. ANODIZE PER MIL-A-8625C, TYPE 2, CLASS I

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED

UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	(STOCK, SIZE, MATERIAL, SPECIFICATION, TYPE, CLASS, GRADE, HEAT TREAT) NOTE TEXT
			-1		ALUMINUM, 606 -T6

PARTS LIST

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APPLICATION

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TOLERANCES

BASIC DIMENSION	DECIMALS 2 PLACE	DECIMALS 3 PLACE	FRACTIONS
UNDER 6	±.01	±.005	±1/32
6-24 INCL.	±.03	±.010	±1/16
OVER 24	±.06	±.015	±1/8
ANGLES	± 0°-30'		

COML TOL APPLY TO STOCK SIZES

CONTRACT		
DRN	E. C. SCHROEDER	5-5-90
CHK	E. C. SCHROEDER	9-23-90
MECH.		
ELEC.		
PROJ.		

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NUT, INLET VALVE RETAINER

SIZE	CODE IDENT. NO.	DRAWING NO.
B	26401	2529029

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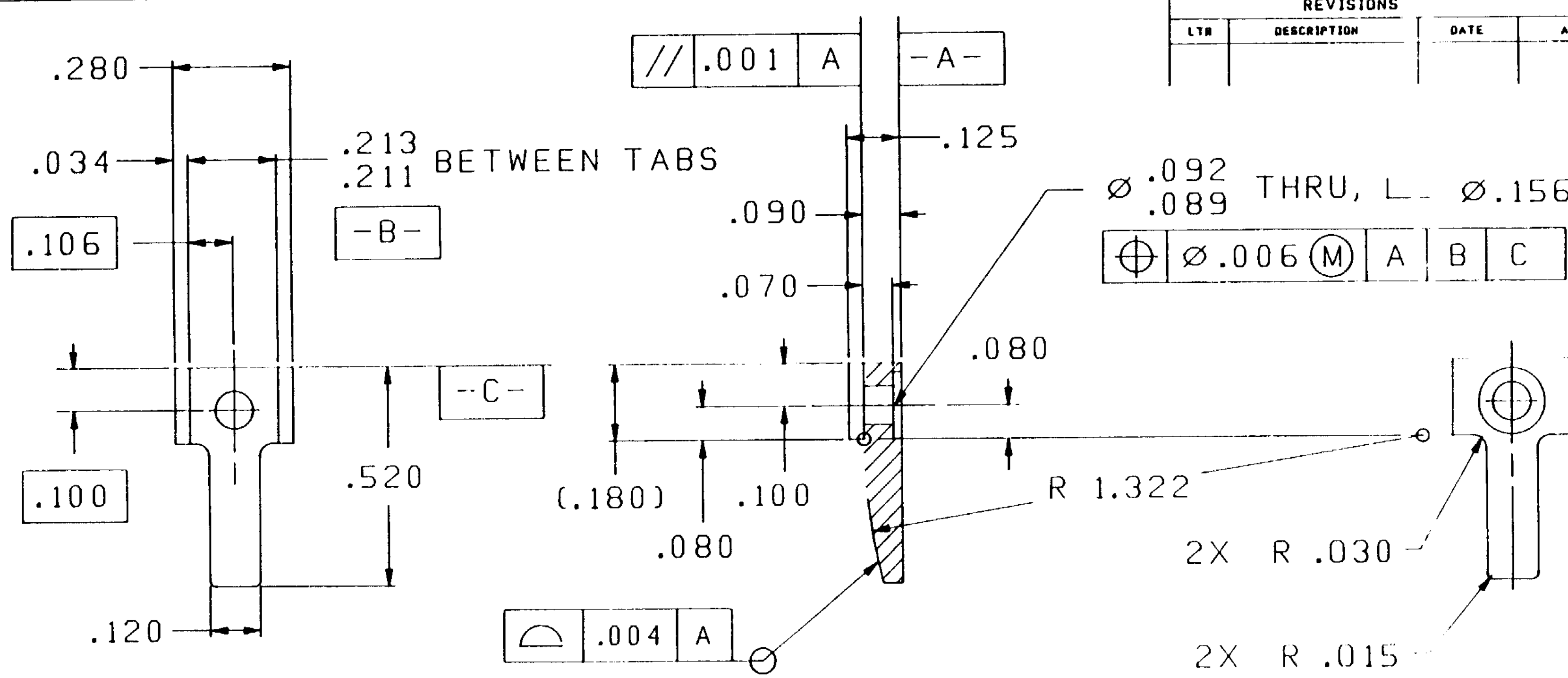

NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV BURRS AND SHARP EDGES EQUIV TO R .005
3. SURFACE FINISH 63 RMS
4. ANODIZE PER MIL-A-8625C, TYPE 2, CLASS I

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED

			-1		ALUMINUM, 6061-T6
UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	(STOCK,SIZE,MATERIAL,SPECIFICATION,TYPE,CLASS,GRADE,HEAT TREAT) NOTE TEXT

PARTS LIST

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TOLERANCES

BASIC DIMENSION	DECIMALS 2 PLACE	DECIMALS 3 PLACE	FRACTIONS
UNDER 6	±.01	±.005	±1/32
6-24 INCL.	±.03	±.010	±1/16
OVER 24	±.06	±.015	±1/8
ANGLES	± 0°-30'		

COML TOL APPLY TO STOCK SIZES

CONTRACT		
DRN	E. C. SCHROEDER	5-4-90
CHK	E. C. SCHROEDER	9-23-90
MECH.		
ELEC.		
PROJ.		

SOUTHWEST RESEARCH INSTITUTE

RETAINER, SECOND STAGE OUTLET VALVE

SIZE	CODE IDENT. NO.	DRAWING NO.
B	26401	2529030
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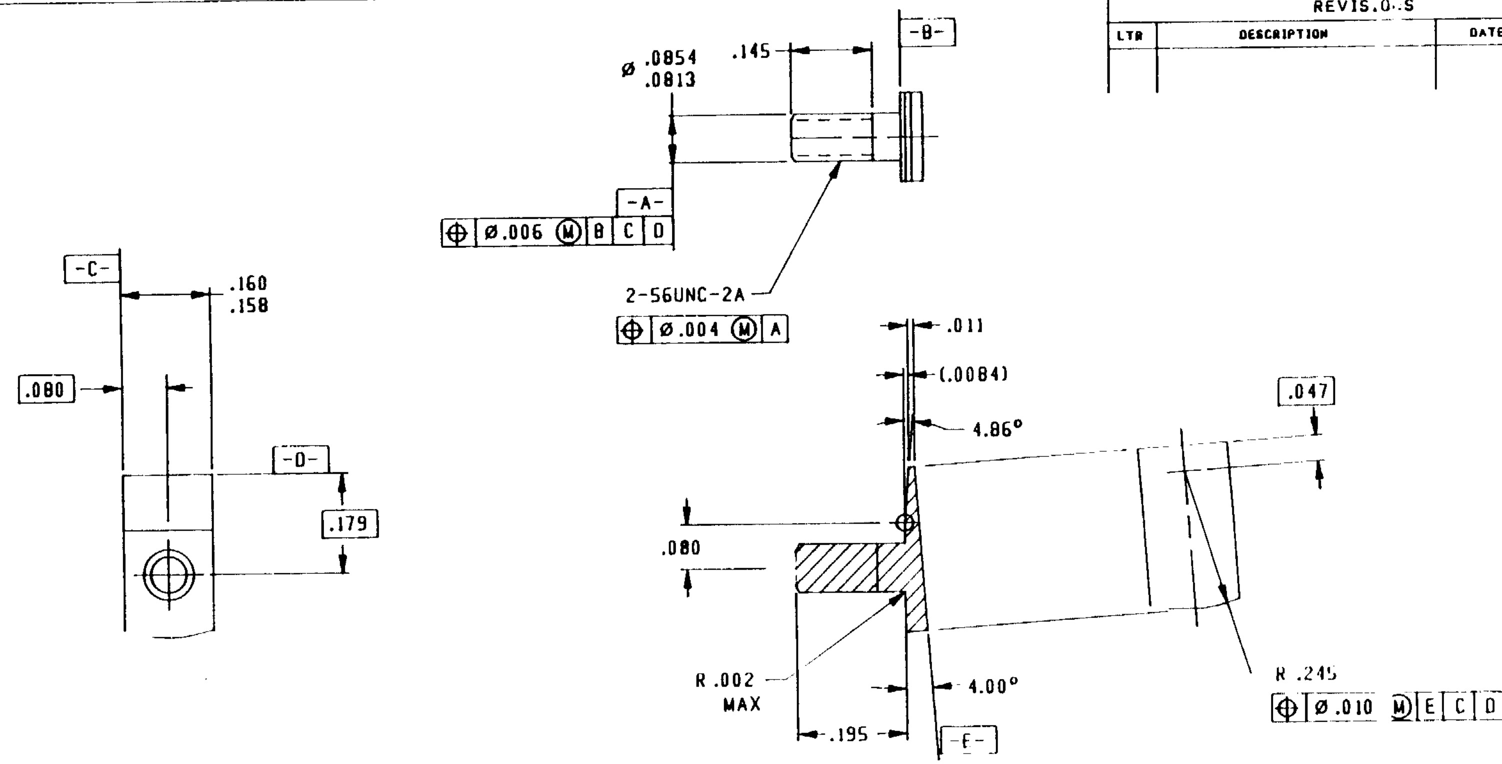

NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV BURRS AND EDGES EQUIV TO R .005
3. SURFACE FINISH 63RMS
4. ANODIZE PER MIL-A-8625C, TYPE 2, CLASS I

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED

			-1		ALUMINUM, 6061-T6
UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	(STOCK,SIZE,MATERIAL,SPECIFICATION,TYPE,CLASS,GRADE,HEAT TREAT) NOTE TEXT

PARTS LIST

NEXT ASSY.	USED ON

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UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES AND INCLUDE CHEMICALLY APPLIED OR PLATED FINISHES

TOLERANCES

BASIC DIMENSION	DECIMALS 2 PLACE	DECIMALS 3 PLACE	FRACTIONS
UNDER 6	±.01	±.005	±1/32
6-24 INCL.	±.03	±.010	±1/16
OVER 24	±.06	±.015	±1/8
ANGLES	± 0°-30'		

COML TOL APPLY TO STOCK SIZES

CONTRACT		
DRN	E. C. SCHROEDER	5-12-90
CHK	E. C. SCHROEDER	9-23-90
MECH.		
ELEC.		
PROJ.		

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RETAINER, SECOND STAGE INTAKE VALVE

SIZE	CODE IDENT. NO.	DRAWING NO.
B	26401	2529031

SCALE 5 - 1 SHEET OF

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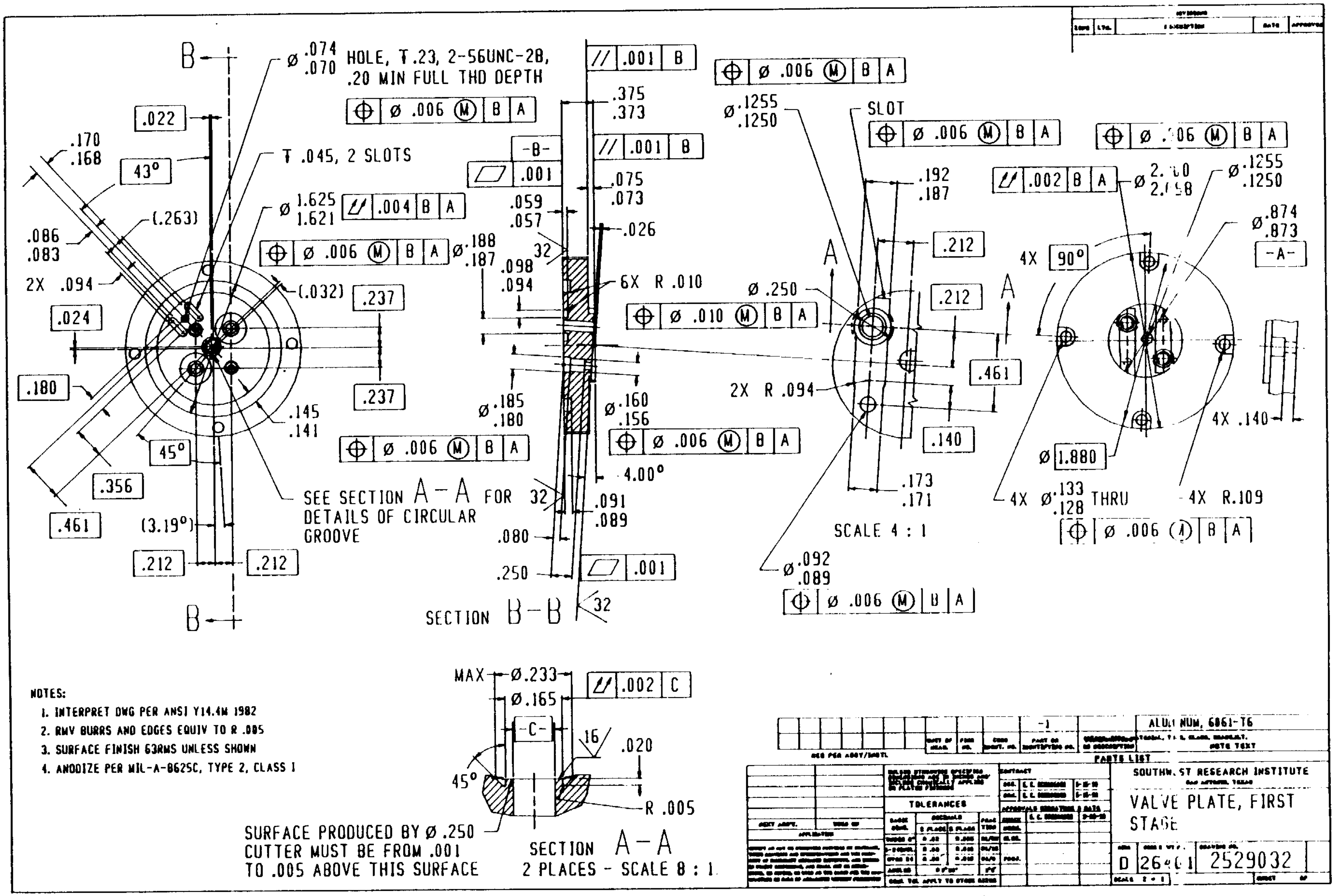

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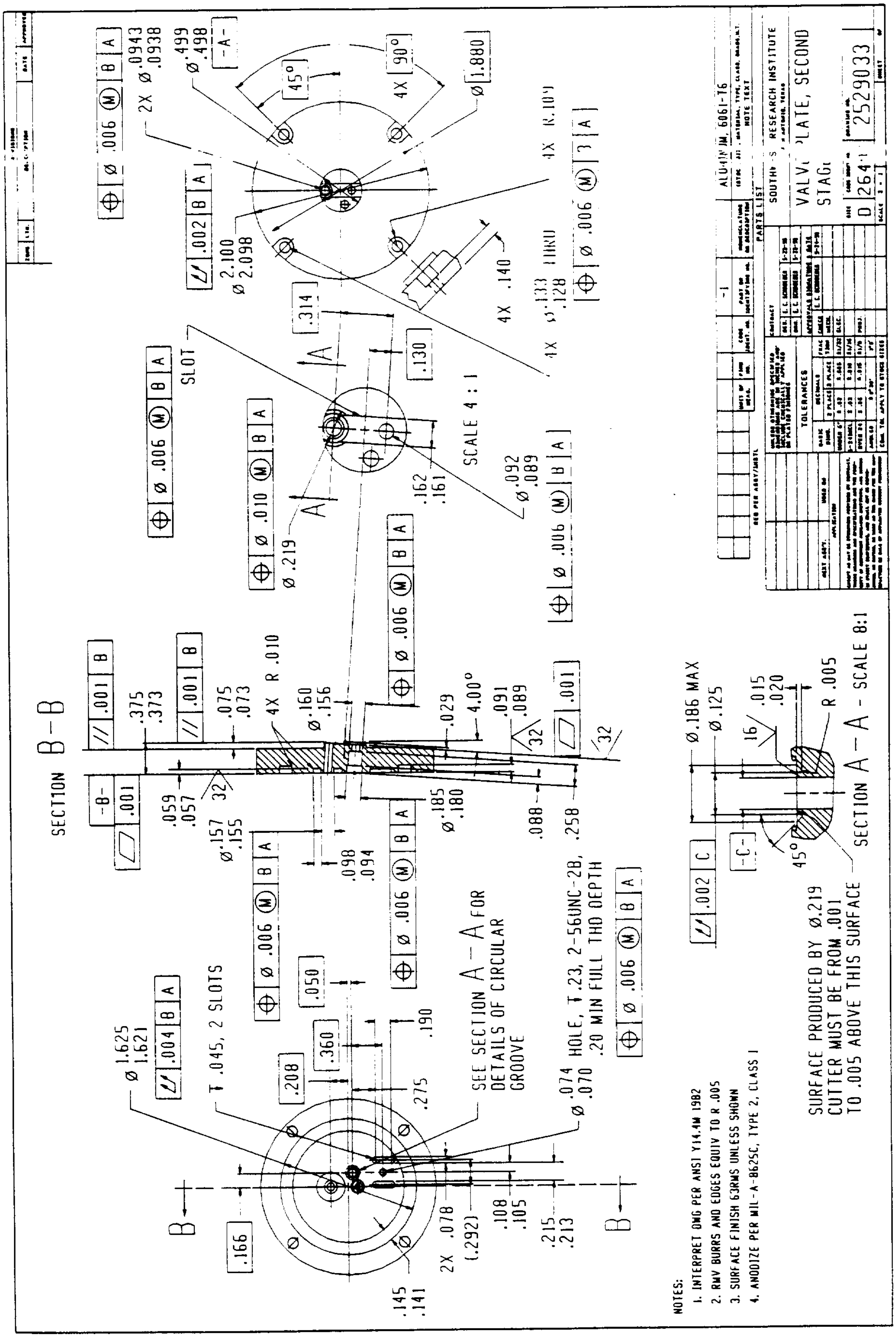

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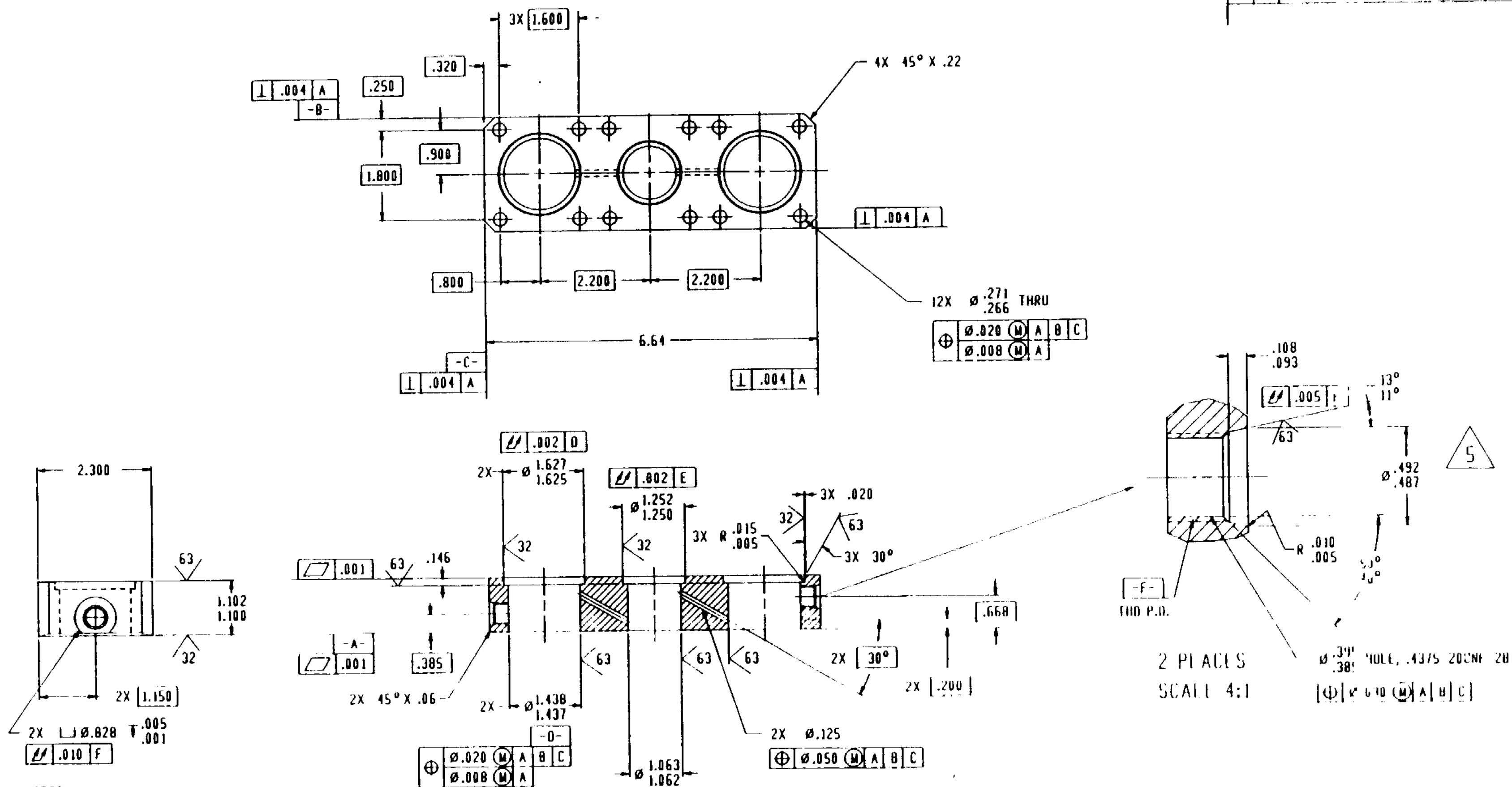

NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND SHARP EDGES EQUIV TO R .005
3. SURFACE FINISH 125RMS UNLESS SHOWN
4. ANODIZE PER MIL-A-8625C, TYPE 2, CLASS I
5. PORT CONTOUR MAY BE MACHINED USING A SIZE 4 SAE FORM TOOL (MS16142), PARKER HANNIFIN CORP PART NO Y34732.

ALUMINUM 6061-T6

PARTS LIST

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MANIFOLD, CYLINDER COOLING FLUID

D 26410 2529034

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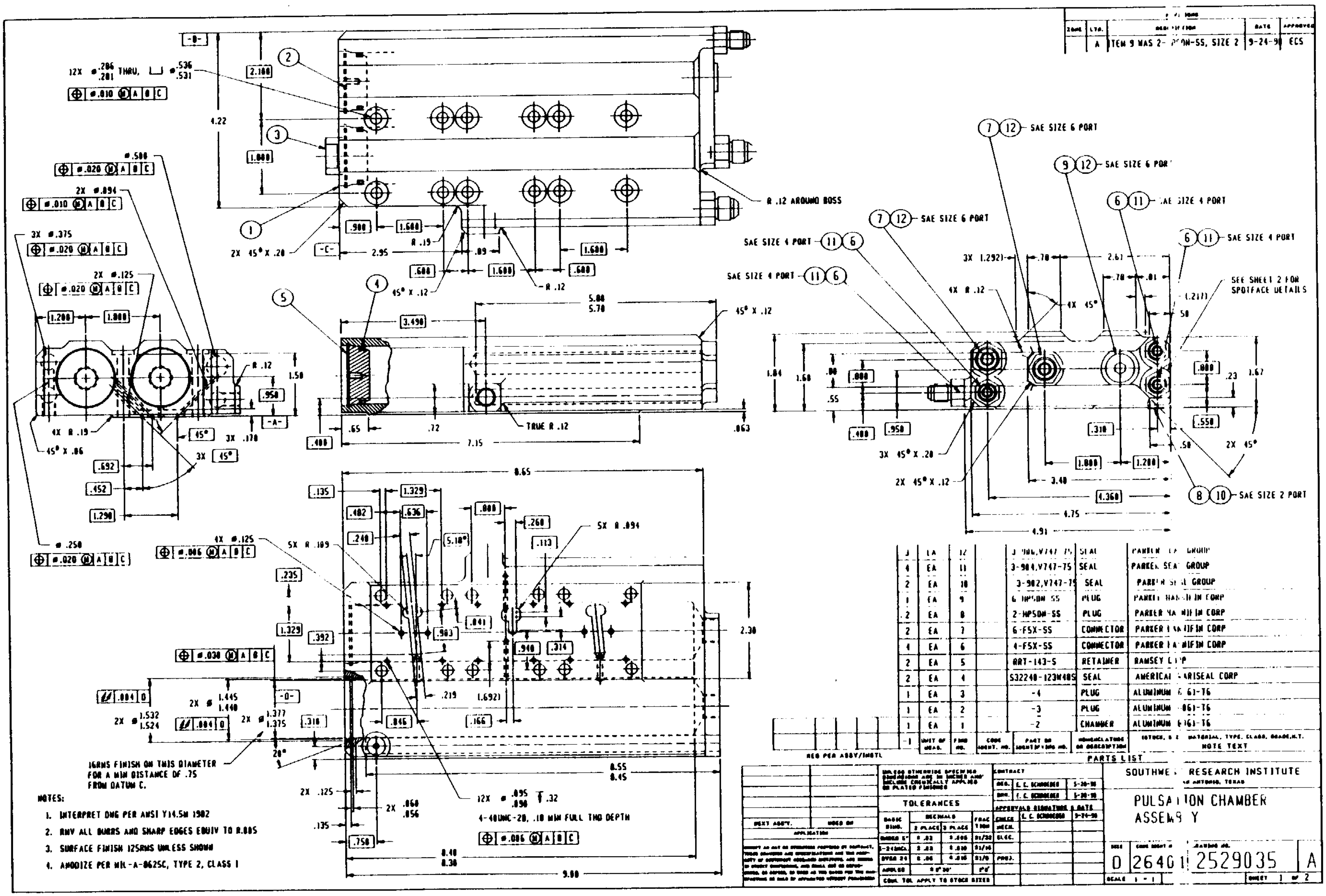
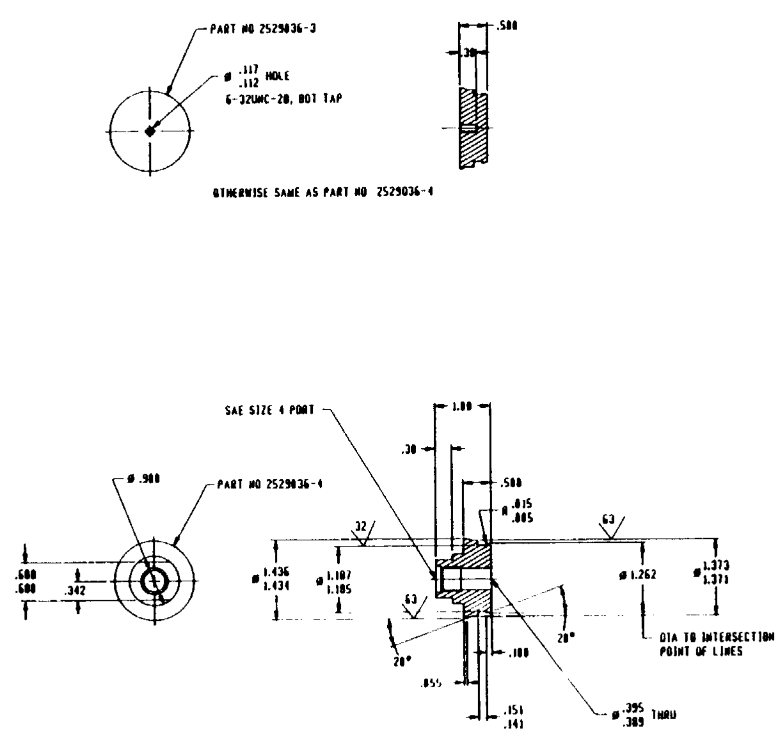
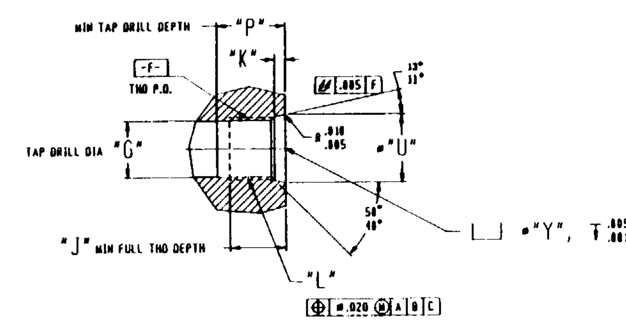


NOTES:
1. INTERPRET DWG PER ANSI Y14.5M 1982
2. RMV ALL BURRS AND SHARP EDGES EQUIV TO R.005
3. SURFACE FINISH 125RMS UNLESS SHOWN
4. ANODIZE PER MIL-A-8625C, TYPE 2, CLASS 1
16RMS FINISH ON THIS DIAMETER FOR A MIN DISTANCE OF .75 FROM DATUM C.
SEE SHEET 2 FOR SPOTFACE DETAILS
R .12 AROUND BOSS
SAE SIZE 6 PORT
SAE SIZE 4 PORT
SAE SIZE 2 PORT
TRUE R .12
4-40UNC-2B, .10 MIN FULL THD DEPTH
A ITEM 9 WAS 2-HP5ON-SS, SIZE 2 9-24-90 ECS
4 EA 11 3-904,V747-75 SEAL PARKER SEAL GROUP
2 EA 10 3-902,V747-75 SEAL PARKER SEAL GROUP
1 EA 9 6-HP5ON-SS PLUG PARKER HANNIFIN CORP
2 EA 8 2-HP5ON-SS PLUG PARKER HANNIFIN CORP
2 EA 7 6-F5X-SS CONNECTOR PARKER HANNIFIN CORP
4 EA 6 4-F5X-SS CONNECTOR PARKER HANNIFIN CORP
2 EA 5 RRT-143-S RETAINER RAMSEY CORP
2 EA 4 S32240-123W40S SEAL AMERICAN VARISEAL CORP
1 EA 3 -4 PLUG ALUMINUM 6061-T6
1 EA 2 -3 PLUG ALUMINUM 6061-T6
1 EA 1 -2 CHAMBER ALUMINUM 6061-T6
PARTS LIST
TOLERANCES
SOUTHWE RESEARCH INSTITUTE
SAN ANTONIO, TEXAS
PULSATION CHAMBER
ASSEMBLY
D 26401 2529035 A
SCALE 1:1 SHEET 1 OF 2



PLUG DETAILS



SAE PORT SIZE	DIMENSION						FORM TOOL PART NO◆
	"G"	"J"	"K"	"L"	"P"	"Y"	
2	.277 .272	.390	.089 .074	.3125-24UNF	.468	.672	Y 4730
4	.395 .389	.454	.108 .093	.4375-20UNF	.547	.828	Y 4732
6	.515 .509	.500	.112 .097	.5625-18UNF	.609	.969	Y 4734

◆ PARKER HANNIFIN CORP

SAE PORT DETAILS

PARTS LIST

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SAN ANTONIO, TEXAS

PULSATION CHAMBER ASSEMBLY

TOLERANCES

SIZE	CODE IDENT	DRAWING NO.
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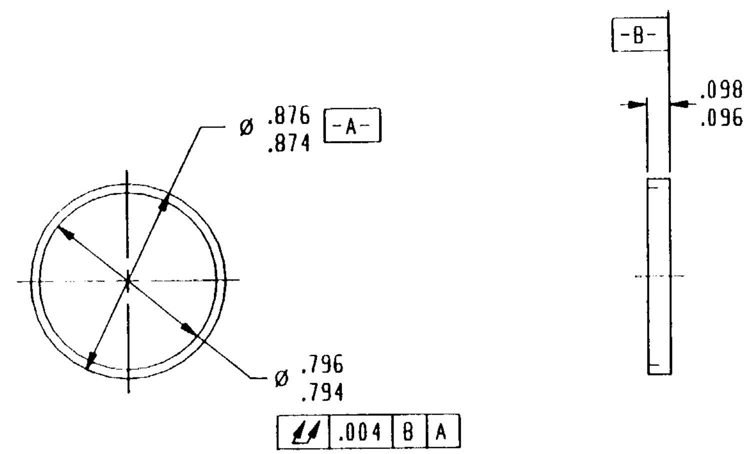
REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED



NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. SURFACE FINISH 63 RMS

UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	(STOCK, SIZE, MATERIAL, SPECIFICATION, TYPE, CLASS, GRADE, HEAT TREAT) NOTE TEXT
			-1		RULON "F", DIXON INDUSTRIES CORPORATION

PARTS LIST

NEXT ASSY.	USED ON

APPLICATION

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UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES AND INCLUDE CHEMICALLY APPLIED OR PLATED FINISHES

TOLERANCES

BASIC DIMENSION	DECIMALS 2 PLACE	DECIMALS 3 PLACE	FRACTIONS
UNDER 6	±.01	±.005	±1/32
6-24 INCL.	±.03	±.010	±1/16
OVER 24	±.06	±.015	±1/8
ANGLES	± 0°-30'		

COML TOL APPLY TO STOCK SIZES

CONTRACT		
DRN	E. C. SCHROEDER	7-4-90
CHK	E. C. SCHROEDER	9-24-90
MECH.		
ELEC.		
PROJ.		

SOUTHWEST RESEARCH INSTITUTE

PISTON RING FIRST STAGE

SIZE	CODE IDENT. NO.	DRAWING NO.
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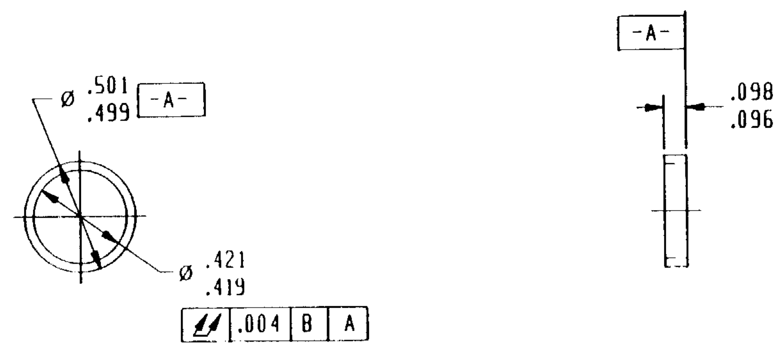
REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED



NOTES:

1. INTERPRET DWG PER ANSI Y14.5M 1982
2. SURFACE FINISH 63 RMS

UNIT OF MEAS.	FIND NO.	CODE IDENT. NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	(STOCK, SIZE, MATERIAL, SPECIFICATION, TYPE, CLASS, GRADE, HEAT TREAT) NOTE TEXT
			-1		RULON "F", DIXON INDUSTRIES CORPORATION

PARTS LIST

NEXT ASSY.	USED ON

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UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES AND INCLUDE CHEMICALLY APPLIED OR PLATED FINISHES

TOLERANCES

BASIC DIMENSION	DECIMALS 2 PLACE	DECIMALS 3 PLACE	FRACTIONS
UNDER 6	± .01	± .005	± 1/32
6-24 INCL.	± .03	± .010	± 1/16
OVER 24	± .06	± .015	± 1/8
ANGLES	± 0°-30'		

COML TOL APPLY TO STOCK SIZES

CONTRACT		
DRN	E. C. SCHROEDER	7-4-90
CHK	E. C. SCHROEDER	9-25-90
MECH.		
ELEC.		
PROJ.		

SOUTHWEST RESEARCH INSTITUTE

PISTON RING, SECOND STAGE

SIZE	CODE IDENT. NO.	DRAWING NO.
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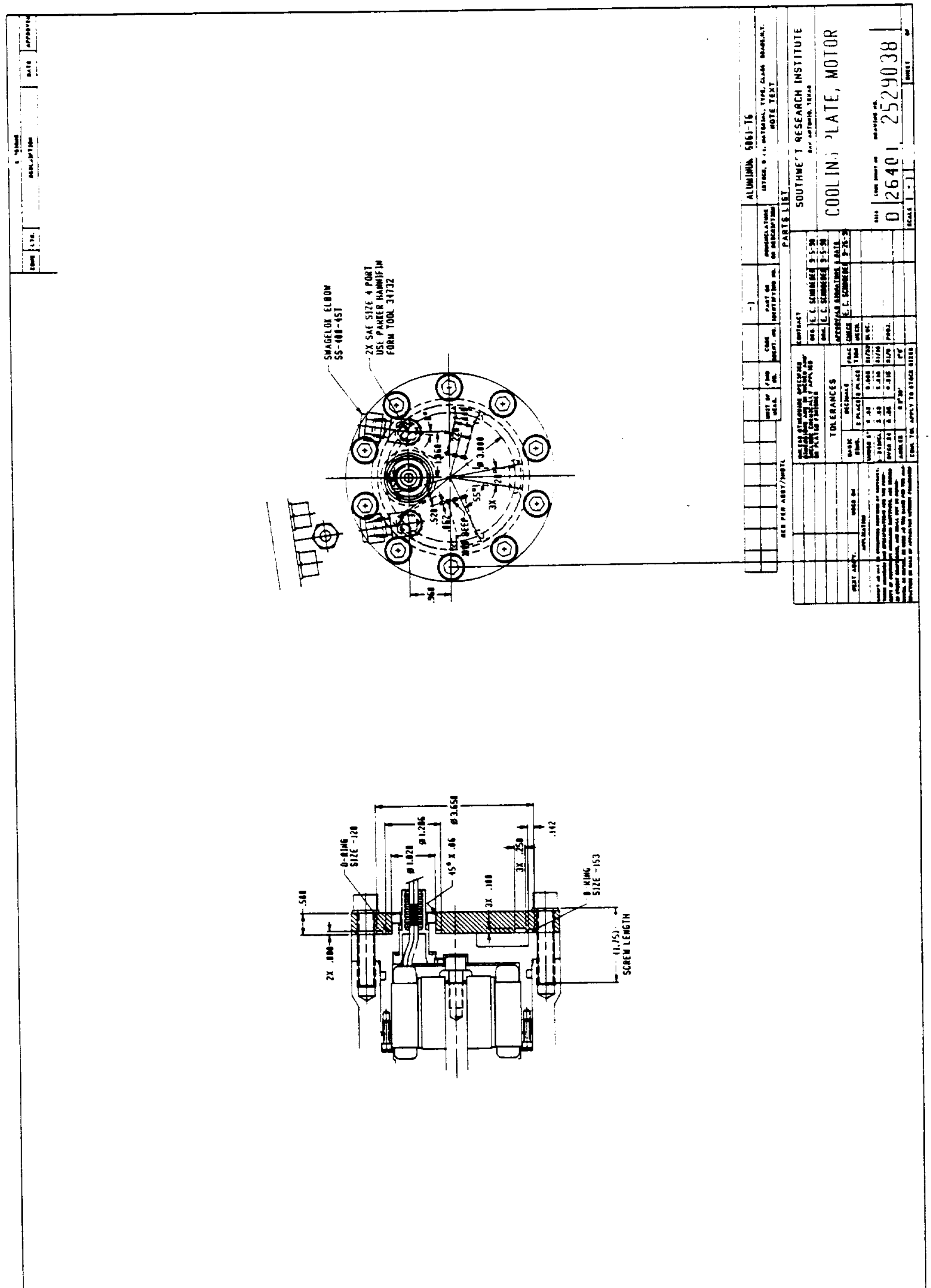

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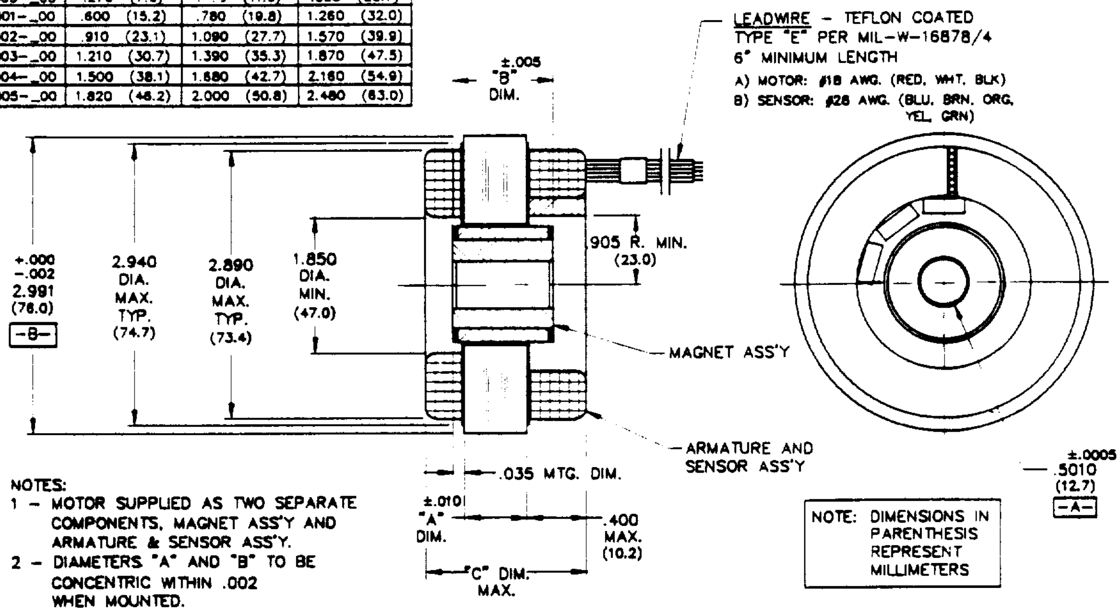
APPENDIX B - MOTOR AND CONTROLLER INFORMATION

Frameless Motor

MODEL	"A" DIM	"B" DIM	"C" DIM
RBE-01800-_00	.275 (7.0)	[illegible] (11.6)	.935 (23.7)
RBE-01801-_00	.600 (15.2)	.780 (19.8)	1.260 (32.0)
RBE-01802-_00	.910 (23.1)	1.090 (27.7)	1.570 (39.9)
RBE-01803-_00	1.210 (30.7)	1.390 (35.3)	1.870 (47.5)
RBE-01804-_00	1.500 (38.1)	1.680 (42.7)	2.160 (54.9)
RBE-01805-_00	1.820 (46.2)	2.000 (50.8)	2.480 (63.0)

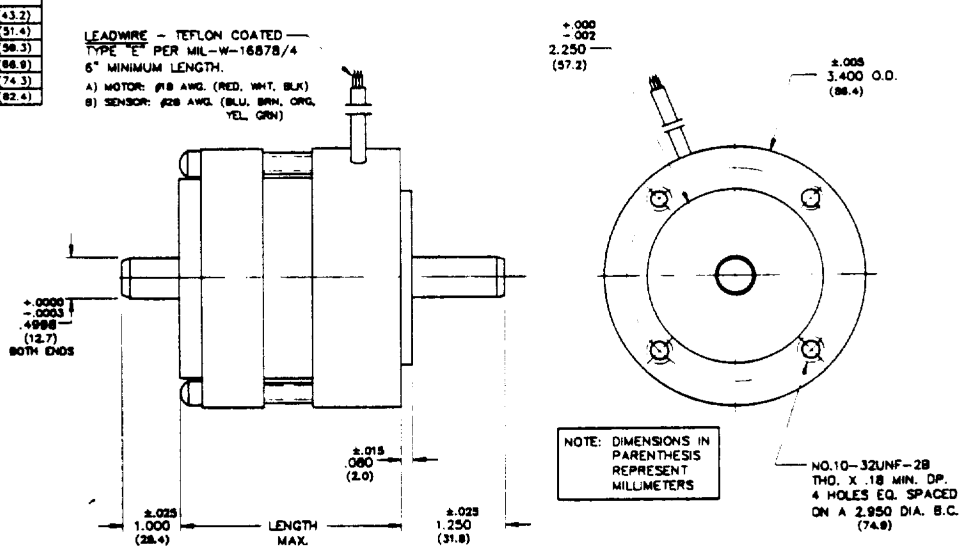


NOTES:
1 - MOTOR SUPPLIED AS TWO SEPARATE COMPONENTS, MAGNET ASS'Y AND ARMATURE & SENSOR ASS'Y.
2 - DIAMETERS "A" AND "B" TO BE CONCENTRIC WITHIN .002 WHEN MOUNTED.
3 - MOUNTING SURFACE BETWEEN 2.991 AND 2.940 DIAMETERS ON BOTH SIDES.

NOTE: DIMENSIONS IN PARENTHESIS REPRESENT MILLIMETERS

Housed Motor

MODEL	LENGTH
RBEH-01800-_00	1.700 (43.2)
RBEH-01801-_00	2.025 (51.4)
RBEH-01802-_00	2.335 (59.3)
RBEH-01803-_00	2.635 (66.9)
RBEH-01804-_00	2.925 (74.3)
RBEH-01805-_00	3.245 (82.4)



NOTE: DIMENSIONS IN PARENTHESIS REPRESENT MILLIMETERS

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SIZE CONSTANTS

PARAMETERS / MODEL NO.		UNITS	RBE-01800	RBE-01801	RBE-01802	RBE-01803	RBE-01804	RBE-01805
Peak Rated Torque, ±25%		oz-in Nm	290 2.0	595 4.2	874 6.2	1130 8.0	1390 [illegible]	1667 11.8
Power at Peak Rated Torque		Watts	609	798	966	1102	1257	1457
Max. Continuous Stall Torque, T_C		oz-in Nm	58 0.41	112 0.79	154 1.09	191 1.35	230 1.62	262 1.85
Max. Continuous Output Power		Watts	102.5	143.7	168.1	186.4	210.1	221.7
Motor Constant, ± 15%, Km		oz-in/$\sqrt{W}$ Nm/$\sqrt{W}$	11.8 0.083	21.2 0.150	28.2 0.199	36.5 0.258	38.1 0.269	43.1 0.304
TPR, ± 15% +		(°C/W)	2.9	2.6	2.4	2.3	2.1	2.0
Viscous Damping, F_I		oz-in-/RPM Nm/RPM	1.1×10^{-3} 7.8×10^{-4}	2.1×10^{-3} 1.5×10^{-3}	2.9×10^{-3} 2.0×10^{-3}	3.7×10^{-3} 2.6×10^{-3}	4.5×10^{-3} 3.2×10^{-3}	5.3×10^{-3} 3.7×10^{-3}
Hysteresis Drag Torque, T_F		oz-in Nm	1.46 0.010	2.86 0.020	4.10 0.029	5.18 0.036	6.34 0.045	7.54 0.053
Max. Cogging Torque		oz-in Nm	3.0 0.021	4.0 0.028	5.3 0.037	7.0 0.049	8.0 0.057	9.0 0.064
Frameless Motor	Inertia, J_M	oz-in-sec² Kg-m²	5.1×10^{-3} 3.6×10^{-6}	8.7×10^{-3} 6.1×10^{-5}	12.2×10^{-3} 8.6×10^{-5}	15.5×10^{-3} 10.9×10^{-5}	18.8×10^{-3} 13.3×10^{-5}	22.3×10^{-3} 15.7×10^{-5}
	Weight	oz gm	9.4 266	17.8 505	25.6 726	33.0 936	40.0 1134	48.0 1361
Housed Motor	Inertia, J_M	oz-in-sec² Kg-m²	5.2×10^{-3} 3.7×10^{-5}	8.8×10^{-3} 6.2×10^{-5}	12.4×10^{-3} 8.7×10^{-5}	15.8×10^{-3} 11.1×10^{-5}	19.1×10^{-3} 13.5×10^{-5}	22.7×10^{-3} 16.0×10^{-5}
	Weight	oz gm	27.0 765	35.2 998	42.9 1216	50.6 1434	57.8 1638	65.7 1862
No. of Poles			12	12	12	12	12	12

24 VOLT 'A' WINDING CONSTANTS

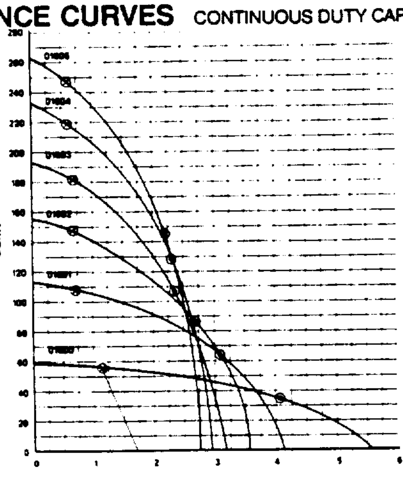
Alternate Windings Available

Peak Torque, ± 25%, T_P		oz-in Nm	183 1.29	360 2.54	552 3.90	730 5.16	873 6.17	1061 7.49
Peak Current, ± 15%, I_P		Amps	10.0	12.0	16.0	20.0	21.8	25.3
Torque Sensitivity, ± 10%, K_T		oz-in/Amp Nm/Amp	18.3 0.129	30.0 0.212	34.5 0.244	36.5 0.258	40.0 0.282	42.0 0.297
No Load Speed, ± 10%		RPM	1700	1030	900	850	780	740
Voltage Constant, ± 10%, K_B		V/Rad/sec V/KRPM	0.129 13.53	0.212 22.18	0.244 25.50	0.258 26.98	0.282 29.57	0.297 31.05
Terminal Resistance, ± 12%, R_M		ohms @ 25°C	2.4	2.0	1.5	1.2	1.1	0.95
Terminal Inductance, ± 30%, L_M		mH	2.4	2.3	2.3	1.7	1.7	1.5
Max. Continuous Output Power	Power	Watts	48.6	58.5	72.5	87.8	96.8	106.3
	Torque	oz-in Nm	55.1 0.39	106.7 0.75	146.1 1.03	179.3 1.26	215.9 1.53	245.1 1.73
	Speed	RPM	1192	740	670	662	606	586

+TPR assumes housed motor mounted to 4.5 x 4.5 x .25" aluminum heat sink or equivalent.

PERFORMANCE CURVES

CONTINUOUS DUTY CAPABILITY FOR 75°C RISE



Design Features of RBE(H) Brushless Motors

- High torque to weight and inertia ratios
- Samarium cobalt rare earth magnets
- 3 phase delta or wye connection
- Housed or frameless designs
- Stationary outer stator winding rotating inner permanent magnet rotor
- Stainless steel shafts (housed versions)
- All motors built to MIL-Q-9858A
- Encapsulated windings available for harsh environments
- Built-in Hall effects for electronic commutation

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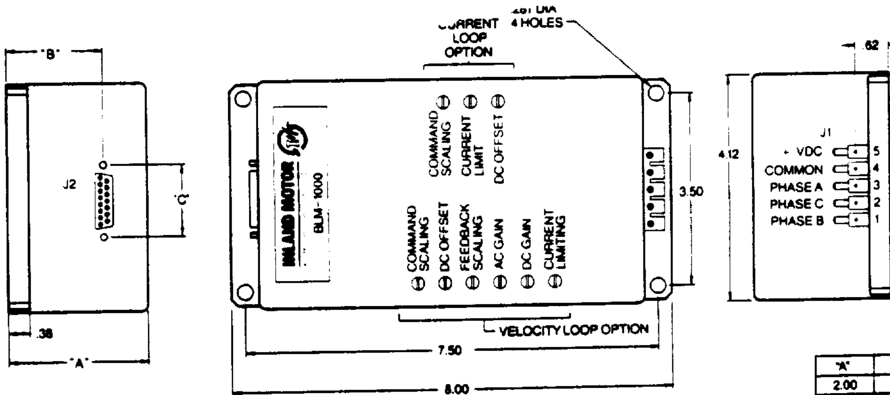
VOLTAGE & CURRENT RATING COMBINATIONS

A. 28 volts 20 amps
B. 70 volts 12 amps

OUTLINE



"A"	"B"	"C"	"J2"	OPTION
2.00	1.11	1.312	15 PIN	CURRENT LOOP
2.50	1.72	1.852	25 PIN	VELOCITY LOOP

FEATURES

- Current Loop Operation
- Velocity Loop Operation with Tachometer, Hall Devices, or Encoder
- Frequency Locked Loop Operation
- 20 KHz PWM Frequency
- EMI, RFI Environment Protection
- Optically Isolated Enable/Reset Line
- Modular Package Size: 8" x 4" x 2"
- Adjustable Current Limit
- Complete Short-circuit Protection
- Baseplate Conduction Cooling
- Greater than 90% Efficient
- Four Quadrant Operation

SPECIFICATIONS

POWER OUTPUT

	A 28 volts/20 amps	B 70 volts/12 amps	C	D	E
MAX VOLTS	40	90			
AMPS CONT.	20	12			
AMPS PEAK	20	12			
WATTS CONT.	800	1080			
WATTS PEAK	800	1080			

POWER INPUT

BUS VOLTAGE	5-45 VDC	5-95 VDC			
CURRENT	0-20 A	0-12 A			
CONTROL VOLTAGE	20-32 VDC	20-30 VDC			
CURRENT	400 mA	400 mA			

LOAD

MIN. INDUCTANCE	1 mH	1 mH			

MECHANICAL

SIZE	8" x 4" x 2"	8" x 4" x 2"			
WEIGHT	2.6 lb	2.6 lb			
SIGNAL CONNECTOR	DBM 15P	DBM 15P			
POWER CONNECTOR	Term. Strip	Term. Strip			

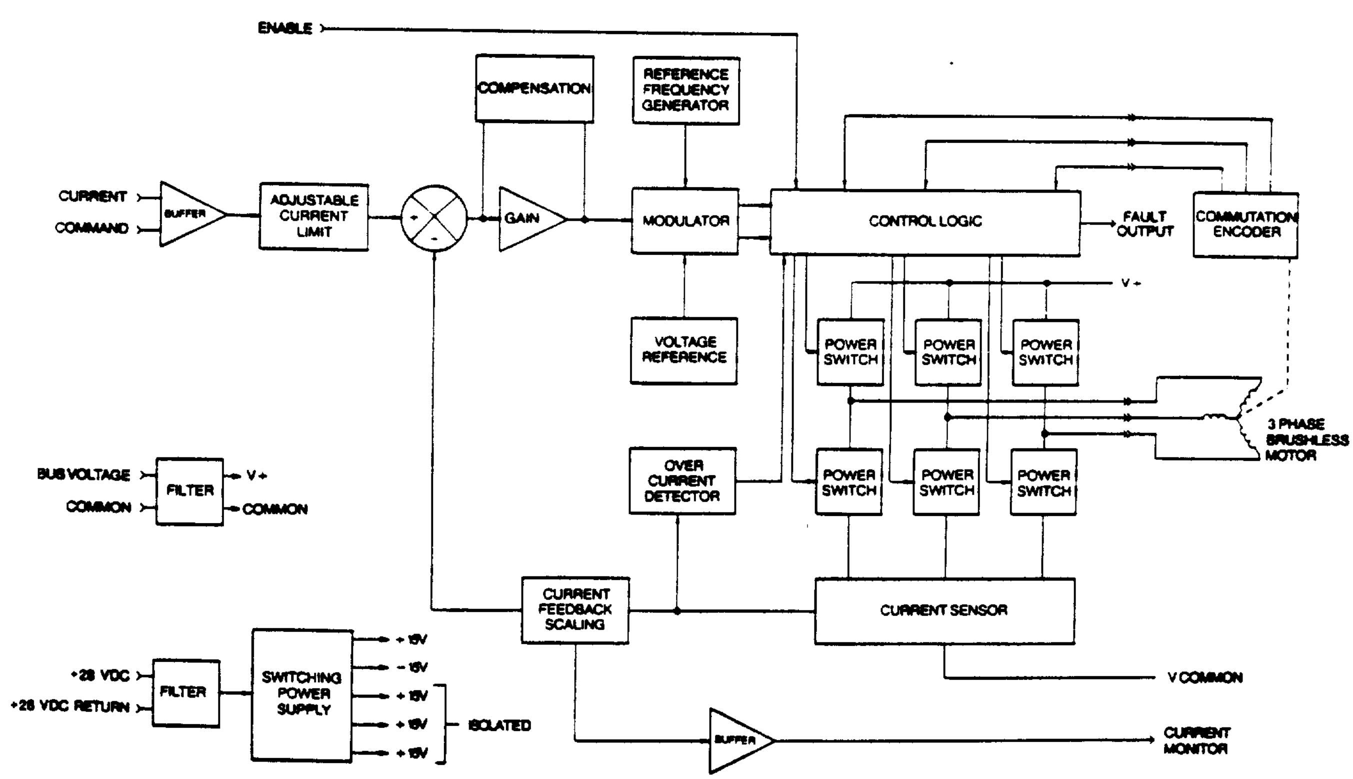
COMMUTATION: Six Sequence

CONTROL CONFIGURATIONS

	COMMAND INPUT	EXTERNAL FEEDBACK	ADJUSTMENTS
CURRENT LOOP	±10V	(NONE)	Command Scaling Current Limit DC Offset
VELOCITY LOOP (BRUSH TACH)	±10V	Brush Tach	Command Scaling, DC Offset Feedback Scaling, AC Gain DC Gain, Current Limit
VELOCITY LOOP * (BRUSHLESS TACH)	±10V	Brushless Tach	Command Scaling, DC Offset Feedback Scaling, AC Gain DC Gain, Current Limit
VELOCITY LOOP* (ENCODER)	±10V	Encoder	Command Scaling, DC Offset Feedback Scaling, AC Gain DC Gain, Current Limit
VELOCITY LOOP* (HALL SENSORS)	±10V	Hall Sensors	Command Scaling, DC Offset Feedback Scaling, AC Gain DC Gain, Current Limit
FREQUENCY LOCKED VELOCITY LOOP*	Ref. Freq.	Hall Sensors Or Encoder	(Factory Pre-set)

* Consult Factory For These Options

BLOCK DIAGRAM OF SERVO DRIVE SYSTEM

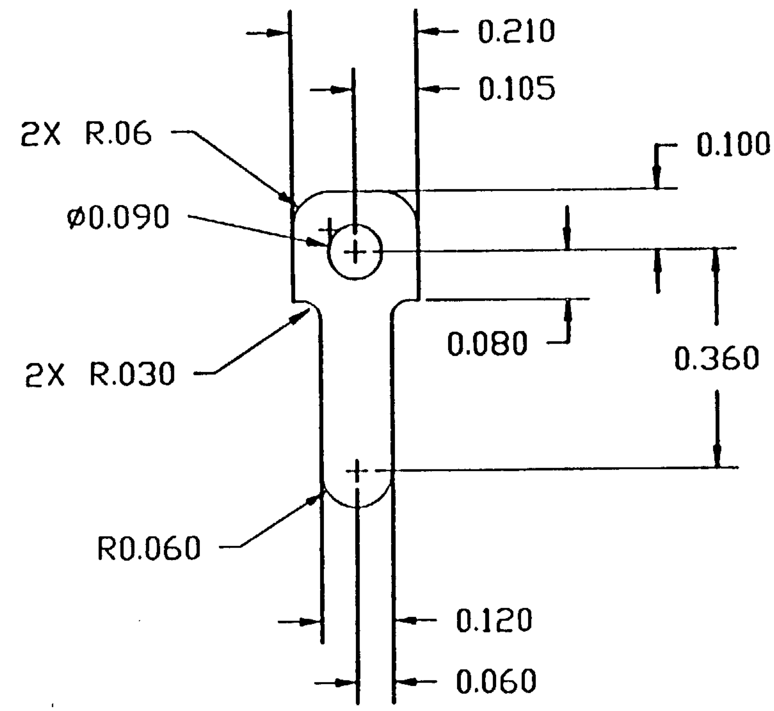


APPENDIX C - CHECK VALVE SPECIFICATION

CHECK VALVE SPECIFICATION

- Pressure Actuated Design
- Material - 302SS

	First Stage	Second Stage
Suction Valve Width (in)	0.170	0.120
Suction Valve Length (in)	0.350	0.200
Discharge Valve Width (in)	0.170	0.120
Discharge Valve Length (in)	0.350	0.250
Suction Valve Thickness (in)	0.004	0.003
Discharge Valve Thickness (in)	0.006	0.004



APPENDIX C

VERIFICATION PROGRAM REPORT FOR ON-ORBIT COMPRESSOR TECHNOLOGY PROGRAM

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

1.0 INTRODUCTION

The Verification Program Report describes the methods used to verify that the compressor meets the design requirements outlined in the End Item Specification (EIS) Sections 3.0 and 4.2. The Verification Program is broken into three sections based upon the verification method employed. The first section describes the verification of EIS items by development testing. The second section consists of verification by analysis, and the third section consists of verification by assessment. The following section presents a brief outline of the compressor design.

2.0 COMPRESSOR DESCRIPTION

The basic design, shown in Figure 1, is a 3-cylinder, two stage reciprocating piston type compressor with pressure actuated check valves. The two outer pistons are the First Stage and the smaller center piston is the Second Stage. The pistons are follower actuated by eccentrically mounted anti-friction bearings. The piston is held in contact with the actuator with a preloaded spring. The following list presents the prototype compressor design parameters:

Table I. Prototype Compressor Design Parameters

	First Stage	Second Stage
Number of Cylinders	2	1
Cylinder Bore (inches)	0.875	0.500
Compressor Nominal Speed (RPM)	650-1000	650-1000
Piston Displacement (cu. inches/cylinder)	0.288	0.049
Stroke (inches)	0.48	0.25
Clearance Volume (%)	6	10
Number of Suction Valves	2	1
Diameter of Suction Ports (inches)	0.125	0.094
Number of Discharge Ports	1	1
Diameter of Discharge Ports (inches)	0.125	0.094
Piston Guide Bore (inches)	1.250	1.250
Return Spring Preload (LB)	15	15
Return Spring Rate (LB/inch)	60	60
Motor Peak Rated Torque (oz.-in.)	400	
Motor Power at Rated Peak Torque (watts)	510	
Maximum Continuous Output Power (watts)	560	

The above outlined design was based on a number of competing design requirements and represents a reasonable trade-off between performance, reliability and life requirements. The design

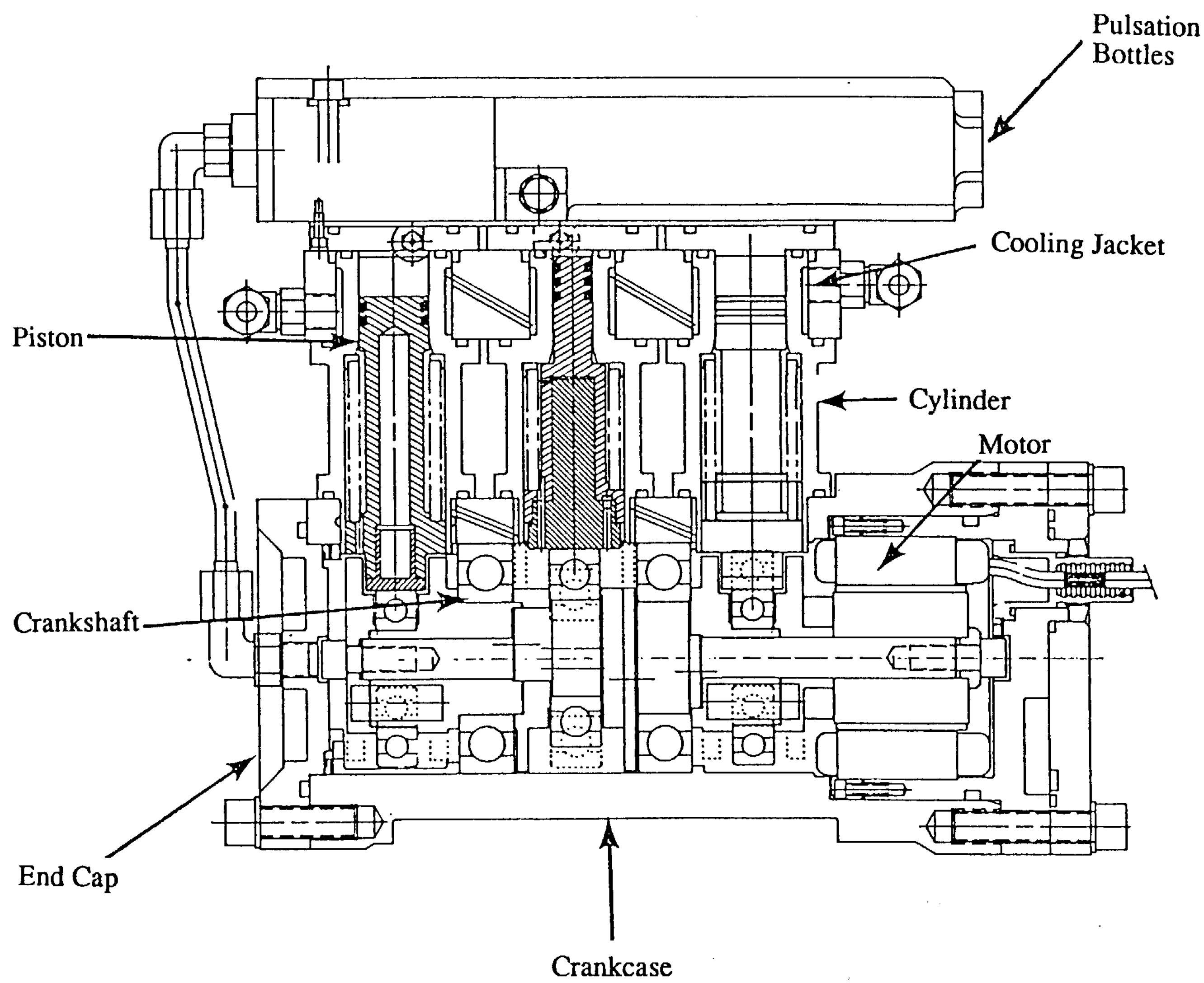


Figure 1. Prototype Compressor Layout

is simple with few moving parts, and based on component wear and life predictions, is free of sudden catastrophic failure. Both compressor stages are driven from a single drive motor and crank assembly resulting in fewer mechanical components and lighter weight compared with separate stages. The integration of both stages also simplifies installation since manifolding between the two stages and the pulsation bottles is incorporated into a single head assembly.

The three cylinder design can be balanced to eliminate primary and residual secondary shaking forces. While the three cylinder design is somewhat more complicated than other possible designs, the ability to limit shaking forces is very important. The option of an unbalanced compressor with compensating hardware (active or passive devices) was investigated and determined to be unacceptable for a variable speed compressor.

3.0 DEVELOPMENT TEST PLAN

In order to ensure that the compressor meets the design requirements in the EIS, some detailed testing of compressor components and the compressor assembly is necessary. Development testing is done to substantiate designs, measure performance, and assure the design is suitable for initiation of formal flight hardware development. Since development testing is not intended to provide flight certification, the formal requirements of controlled design, formal certification, formal retest, and flight type hardware are not required.

The EIS contains design requirements common to all compressor applications in Section 3.0 of the EIS and the design requirements specific to Type II compressor in Section 4.2 of the EIS. This portion of the Verification Plan contains only those items in EIS Sections 3.0 and 4.2 that require testing to verify the design requirements are met. The EIS items not requiring verification by testing are verified by analysis or assessment and are discussed in Sections 4.0 and 5.0 of this document.

A description of the test objective and test plan for each EIS item to be verified by test is listed in Table II.

3.1 Strength Testing

3.1.1 Design Requirement: EIS 3.2.2.1 Strength

The components shall have sufficient strength at the design temperature to withstand both limit loads and pressures without loss of operational capability for the life of the component, and the proof loads and pressures at the design temperatures without functional failure during testing.

Table II. Verification By Test

EIS Section No. & Title		Method of Verification
3.2.2.1	Strength	Exempt Except For Proof Pressure - Test
3.2.2.3	Surface Wear	Analysis & Test
3.2.2.5	Weight	Test
3.2.2.6	Envelope	Test
3.3.1.13	Cleanliness Verification	Test
3.3.1.20	Surface Texture	Assessment & Test
4.2.2.1	Operating Pressures	Test
4.2.2.2	Proof Pressure	Test
4.2.3	Fluid Operating Temperatures	Test
4.2.4	Fluid Flow Rate	Test
4.2.5.1	Operating Life	Test & Analysis
4.2.6	Power Limitations	Test

3.1.2 Test Item

Prototype compressor assembly is described in the Prototype Final Design Report.

3.1.3 Test Description

The compressor assembly was subjected to the proof pressure of 10.35 MPa (1500 psi) for a period of five minutes. During the application of the proof pressure the compressor was not operating. The proof pressure was applied at the suction and discharge ports of the compressor with the compressor at room temperature (75 ±5°F). The test gas was nitrogen.

After the proof pressure was relieved, the compressor performance was verified to ensure the performance specifications outlined under EIS 4.2 were still met. The testing was performed on the test assembly and under the same operating conditions outlined under EIS 4.2.

3.1.4 Test Equipment

Pressure Gage—for measuring proof pressure.

Manufacturer: Wika Instruments Corp.

Model: 232.33

Range: 0 - 1500 psig

Calibration Points—0%, 25%, 50%, 75%, and 100% of Full Scale

Dead Weight Pressure Calibrator—for pressure gage calibration.

Manufacturer: Ashcroft (Dresser)

Model: Dead Weight Tester 2HH-286SI

Range: 0 - 10,000 psi

3.1.5 Test Results

Prior to applying proof pressure, a leak check with soap solution revealed a minor leak in the case vent fitting that was corrected by tightening the fitting. After the unit was free of leakage, the five minute proof test was initiated. No problems occurred during the proof test. After the proof pressure was relieved, the compressor performance was tested at 650 RPM and seven different flow rates. The results of the performance test after the application of the proof pressure were the same as the performance testing prior to proof testing. The compressor passed the proof test without any degradation in performance, leakage, or breakage.

3.2 Surface Wear Testing

3.2.1 Design Requirement: EIS 3.2.2.3 Surface Wear

The wear and attendant particle generation at any dynamically interfacing surfaces (contacting surfaces under relative motion) shall not introduce contaminant into the fluid flow path and shall not impair the function of that specific interface, the compressor as a whole, or the user system for the life of the compressor.

3.2.2 Test Item

Subassembly Test Article (STA) is described in the Prototype Final Design Report.

3.2.3 Test Description

The surfaces subject to wear in the compressor are the piston seals and guide rings, the outer bearing race acting on the cam follower, the bearing races on the balls, and the check valves motion against the valve seat.

Wear testing was performed on the STA which is a single piston compressor. The STA design is modeled after the prototype compressor and contains a piston seal ring, upper and lower guide rings, eccentrically mounted bearing acting on a cam follower, and the fluid check valves. The primary purpose for wear testing in the STA is to provide wear data on the piston seal. Information on the guide rings, bearing, valves and cam follower were also obtained but only from visible observations of the wear surfaces.

The STA piston seal and guide rings were weighed and dimensions measured prior to assembly to determine starting conditions. The compressor was then assembled and the compressor run for 8 hours at 1000 RPM with the inlet at atmospheric pressure (temperature of 75°F) and the discharge at 100 psig. The test gas was air. At the end of the 8 hours, the compressor was disassembled and the wear surfaces remeasured to record the initial break-in wear rate. The compressor was reassembled and run for an additional 1800 hours. The unit was disassembled weekly to determine the post break-in wear rate as a function of time.

3.2.4 Test Equipment

Scale—for weighing the piston seal and guide rings.

Manufacturer: Mettler

Model: H6T

Range: 0 - 160 grams

Resolution: 0.0001 grams

Calibration: Texas Scales Co. calibration certificate

Micrometer—for measuring piston seal and guide rings.

Manufacturer: Mitutoyo

Model: 293-765 (8116910)

Resolution: 0.00005"

Calibration: standard calibration blocks prior to use

Internal micrometer—for measuring piston bore.

Manufacturer: Mitutoyo

Model: 168-207

Range: 0.8" - 1"

Resolution: 0.0002"

Calibration: ring gage prior to use

Internal micrometer—for measuring piston bore.

Manufacturer: Mitutoyo

Model: 368-204

Range: 0.5" - 0.65"

Resolution: 0.0002"

Calibration: ring gage prior to use

Caliper—for measuring piston OD in seal grooves.

Manufacturer: Mitutoyo

Model: 505-626

Resolution: 0.001"

Calibration: micrometer standards

3.2.5 Test Results

The test results of wear testing on the STA are documented in the Prototype Final Design Report and summarized in Figure 2 (see Section 4.3 for a discussion of the life prediction based on the wear data). Wear and particle generation on the cam follower bearings, and check valves was not measurable and particulate could not be observed. These items completed over 2500 hours of run time while several different piston seals were evaluated. The piston seal wear is the only area of concern for particle generation since it has the potential to migrate with the compressed gas. With the piston seals made of inert Teflon compounds, the particulate migration is of minimal concern. During STA testing, the seal wear products were found in the cylinder, on the valve plate and a very small amount around the discharge valve port. These particles may collect in the pulsation bottle where their transport rate will be greatly reduced because of the very low gas velocity in the pulsation bottle. These wear products will not impair the function of the compressor since the quantity of wear products is very small. A filter installed in the compressor discharge line will prevent the particulate migration into downstream equipment. The wear data presented in Figure 2 is an average of five measurements evenly spread around the circumference of the seal ring. The last data set, however, exhibited a "flat-spot". Upon examination of the test rig the lower guide ring had worn through and the piston was not centered. This failure of the test rig occurred sometime between the 1200 hour point and the 1800 hour point. The cumulative loss from the 1800 hour point is somewhere in the range of 0.0106 inches (average for all five locations) and 0.0085 inches (average for four locations by eliminating the flat-spot). If the test rig failure had not occurred it is our judgement that the actual loss would be between the "worst-case" condition and optimum condition.

3.3 Weight

3.3.1 Design Requirement: EIS 3.2.2.5 Weight

Minimum weight shall be a design objective. The weight shall not exceed 36.3 kg (80 lbs.)

3.3.2 Test Item

Prototype compressor assembly.

3.3.3 Test Description

This test was conducted to verify that the compressor prototype meets the above requirement. The assembly weighed consisted of the compressor prototype, coolant lines, inlet and outlet gas fittings, and the mounting baseplate. This entire assembly was placed on a standard balance scale.

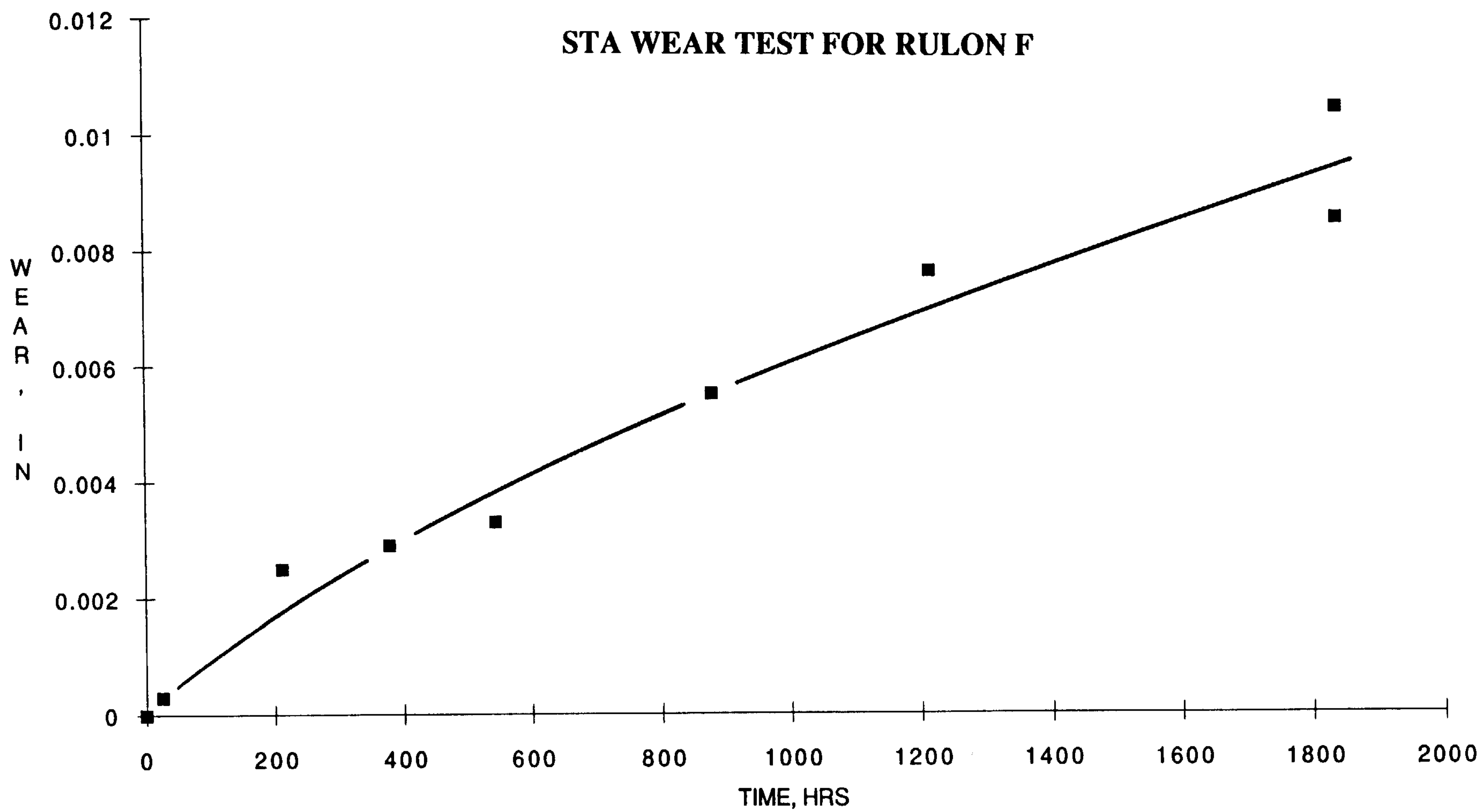


Figure 2. STA Wear Test for Rulon F

3.3.4 Test Equipment

The test equipment is a standard balance beam scale.

Manufacturer: Ohaus

Model: Heavy Duty Solution Balance

Range: 0-45 lbs.

3.3.5 Test Results

The total assembly weight was 30.1 lbs. The assembly minus the coolant lines, fittings, and baseplate weight was 26.9 lbs.

3.4 Envelope Volume

3.4.1 Design Requirement: EIS 3.2.2.6 Envelope

Minimum envelope volume shall be a design objective. The maximum envelope shall be 1.5 cubic feet.

3.4.2 Test Item

Prototype compressor assembly.

3.4.3 Test Description

The overall dimensions of the compressor prototype assembly were measured using a standard mechanical scale. Those overall dimensions included mounting bolts, fitting and electrical wiring such that the entire assembly would fit into an envelope of inner dimensions provided.

3.4.4 Test Equipment

The test equipment is a standard mechanical scale.

3.4.5 Test Results

The overall envelope is:

12.6 inch length

7.0 inch width

9.9 inch height

The overall volume is 873.18 cubic inches or 0.505 cubic feet.

3.5 Cleanliness Verification Testing

3.5.1 Design Requirement: EIS 3.3.1.13 Cleanliness Verification

Following precision cleaning, unless otherwise specified, each item shall be rinsed using 100 milliliters of unused precision cleaning solvent for each square foot of critical surface. Rinsing

shall be accomplished by agitation, sloshing, or by spraying the test solvent over the critical surface in such a manner as necessary to obtain a reliable test solution. The test solvent shall be drained immediately to prevent particle redeposition on the test surface. Particulate determination shall be made in accordance with SEA-ARP-598. NVR determination shall be made in accordance with ASTM D2109-78. Allowables shall be in accordance with EIS 3.3.1.12.

3.5.2 Test Item

Prototype compressor components.

3.5.3 Test Description

Precision cleaning is not required for any surfaces on the compressor. The compressor components will be cleaned in an ultrasonic cleaner immediately prior to assembly.

3.5.4 Test Equipment

None Required.

3.5.5 Test Results

The compressor components were cleaned prior to assembly.

3.6 Surface Texture Measurement

3.6.1 Design Requirement: EIS 3.3.1.20 Surface Texture

Surface texture limitations shall be in accordance with ANSI B46.1-78.

3.6.2 Test Item

Prototype compressor components.

3.6.3 Test Description

Surface texture for critical mating surfaces will be determined in accordance with ANSI B46.1-78.

3.6.4 Test Equipment

None Required.

3.6.5 Test Results

As each compressor component was manufactured, its surface texture was checked to make certain it complied with the design requirements. Because extremely smooth surfaces (< 16 RMS) were not required, machined finishes were verified with visual and tactile comparisons.

3.7 Compressor Performance Testing

3.7.1 Design Requirement: EIS 4.2 Type II Compressor Application Requirements

The following design requirements must be met for the Type II compressor.

The bulk working fluid will consist of an oxidizing gas mixture as specified in Appendix 2, paragraph 1.2 of the EIS. Trace contaminants potentially mixed with the bulk gas mixture are specified in Appendix 2, Section 2.0 of the EIS. Potential phase change or chemical reaction issues that exist with the compression of this gas mixture include, but are not limited to, the following:

(1) condensation of CO_2 at low temperatures (below 0°F and 300 psia)

(2) a possible reaction between fuels (C_2H_2, NH_3, C_2H_4, C_3H_8, etc.) and oxygen at elevated temperatures.

Operating pressures range as follows:

(a) Inlet: 0.07 to 0.20 MPa (10 to 30 psia)

(b) Outlet: 0.69 to 6.9 MPa; 8.28 MPa max (100 to 1000 psia; 1200 psia max).

Proof pressure shall be 1.5 times the operating pressure, approximately 10.35 MPa (1500 psia) as a minimum, for a duration of at least 5 minutes.

Fluid inlet temperature range is as follows:

(a) Maximum: 32.2°C (90°F)

(b) Minimum: 15.5°C (60°F)

Flow rates during compressor operations are as follows:

(a) Nominal: 0.11 Kg/hr (0.25 LBm/hr)

(b) Maximum: 0.50 Kg/hr (1.1 LBm/hr)

3.7.2 Test Item

Prototype compressor assembly.

3.7.3 Test Description

Performance testing on the prototype compressor was conducted to verify the above requirements were met. The tests to be performed encompass the required operating pressures, temperatures, and flow rates. All of the tests were performed with the compressor at room temperature (75°F) and at steady operating conditions. The working fluid was nitrogen.

3.7.4 Test Equipment

Temperature Sensors—for fluid temperature measurements.

Manufacturer: Omega

Model: TMQSS-062U-6

Signal Conditioner Manufacturer: Fluke

Range: 0 - 100°F

Calibration: ASTM Thermometer Set, Ice Point

Pressure Sensors—for gas pressure measurements. (inlet, inner-stage, discharge)

Manufacturer: Wika Instruments Corp.

Model: 232.33

Range: 30 in Hg-0-15 psi, 0 - 300, 0 - 400, 0 - 1500 psig

Calibration: Dead weight tester at 0%, 25%, 50%, 75%, and 100% FS

Dead Weight Pressure Calibrator—for pressure sensor calibration.

Manufacturer: Ashcroft (Dresser)

Model: 10,000 psi Dead Weight Tester, Ser # 2HH-286SI

Multimeter Meter—to measure motor power consumption.

Manufacturer: Hewlett-Packard

Model: 3465A

Shunt Resistor—to measure compressor speed.

Range: 0 - 1000 amps

Resistance: 0.0001 Ω

Flow Meter—to measure fluid flow rate.

Manufacturer: Aalborg Instruments

Model: PRO34/1-082-03C, PRO34/1-102-05C

Range: 0 - 5.18, 0 - 22.8, SCFH

Calibration: Wet Test Meter

Counter—to measure compressor speed.

Manufacturer: Tektronix

Model: DC-505 Universal Counter/Timer

3.7.5 Test Results

To characterize the compressor performance, testing was performed at three different suction pressures. The test results are summarized in Tables III through V and Figures 3 through 5. The tables show the measured parameter such as pressure, flow rate, and temperature. The figures show

Table III. Performance Test Results at 10 psia Suction Pressure

Run #	1	2	3	4	5	6	7	8	9	10	11	12
RPM	659	661	662	663	666	668	1159	1162	1166	1164	1165	1172
Suction Pressure (in Hg)	-9	-9.1	-8.9	-9	-9	-8.9	-9	-8.9	-9.1	-8.9	-9	-9
Innerstage Pressure (psig)	100	73	52	42	17	1	119	94	76	61	44	3
Discharge Pressure (psig)	471	321	190	117	19	0	598	440	330	212	88	0
Suction Temperature (C)	23.3	23.6	23.6	23.6	23.6	23.5	25.3	25.4	25.1	24.9	24.9	24.5
Innerstage Temperature (C)	23.2	23.3	23.4	23.5	23.9	23.7	24.8	24.9	24.9	25.1	25.6	25.6
Discharge Temperature (C)	23.4	23.5	23.6	23.6	23.5	23.4	25.2	25.4	25.3	25.4	25.4	24.9
Flow Rate (Lbm/hr)	0	0.051	0.114	0.164	0.314	0.441	0	0.076	0.148	0.267	0.416	0.815
Power (watts)	360	360	360	360	348	336	432	444	432	420	408	372

Table IV. Performance Test Results at 15 psia Suction Pressure

Run #	5	6	7	8	9	14	15	16	17	18	19
RPM	1158	1158	1158	1159	1173	667	663	665	665	668	675
Suction Pressure (psig)	0.8	0.7	0.7	0.8	0.8	0.6	0.7	0.8	0.9	0.8	0.8
Innerstage Pressure (psig)	184	168	128	100	6	155	124	85	70	36	2
Discharge Pressure (psig)	915	820	515	280	0	680	500	233	131	38	0
Suction Temperature (C)	26.3	26.2	26.1	26	26.9	24.2	24.1	23.9	23.9	23.9	23.5
Innerstage Temperature (C)	25.6	25.7	26.2	26.7	26.1	23.8	23.7	24	24.2	24.7	23.7
Discharge Temperature (C)	26.1	26.3	26.7	26.8	25.3	24.2	24.1	24.1	24.1	24.1	23.4
Flow Rate (Lbm/hr)	0	0.054	0.297	0.537	1.432	0	0.092	0.252	0.344	0.57	0.799
Power (watts)	504	480	492	492	384	408	408	384	396	384	348

Table V. Performance Test Results at 27 psia Suction Pressure

Run #	1	2	3	4	5	16	17	18	19	10	11	12	13	14
RPM	655	656	656	658	659	662	680	1145	1145	1144	1147	1148	1151	1157
Suction Pressure (psig)	12.4	12.2	12.3	12.4	12.3	12.1	12.2	12.4	12.3	12.1	12.2	12.2	12.2	12.2
Innerstage Pressure (psig)	283	249	222	194	165	136	10	285	257	227	198	174	150	120
Discharge Pressure (psig)	1210	1000	800	600	400	202	0	1218	1012	800	570	390	220	120
Suction Temperature (C)	25.2	25.2	25.1	24.8	24.7	24.5	23.6	26.3	26.6	26.6	26.6	26.5	26.3	25.4
Innerstage Temperature (C)	24.7	24.7	24.9	25	25.3	25.5	23.7	26.4	27.1	27.6	28.3	29.1	29.2	28.7
Discharge Temperature (C)	25.1	25.1	25.2	25.2	25.3	25.1	23.8	27	27.5	27.9	28.2	28.4	28.1	26.5
Flow Rate (Lbm/hr)	0	0.124	0.241	0.384	0.52	0.673	1.827	0.241	0.461	0.654	0.885	1.069	1.285	1.567
Power (watts)	480	480	480	468	456	444	324	588	576	576	552	564	528	492

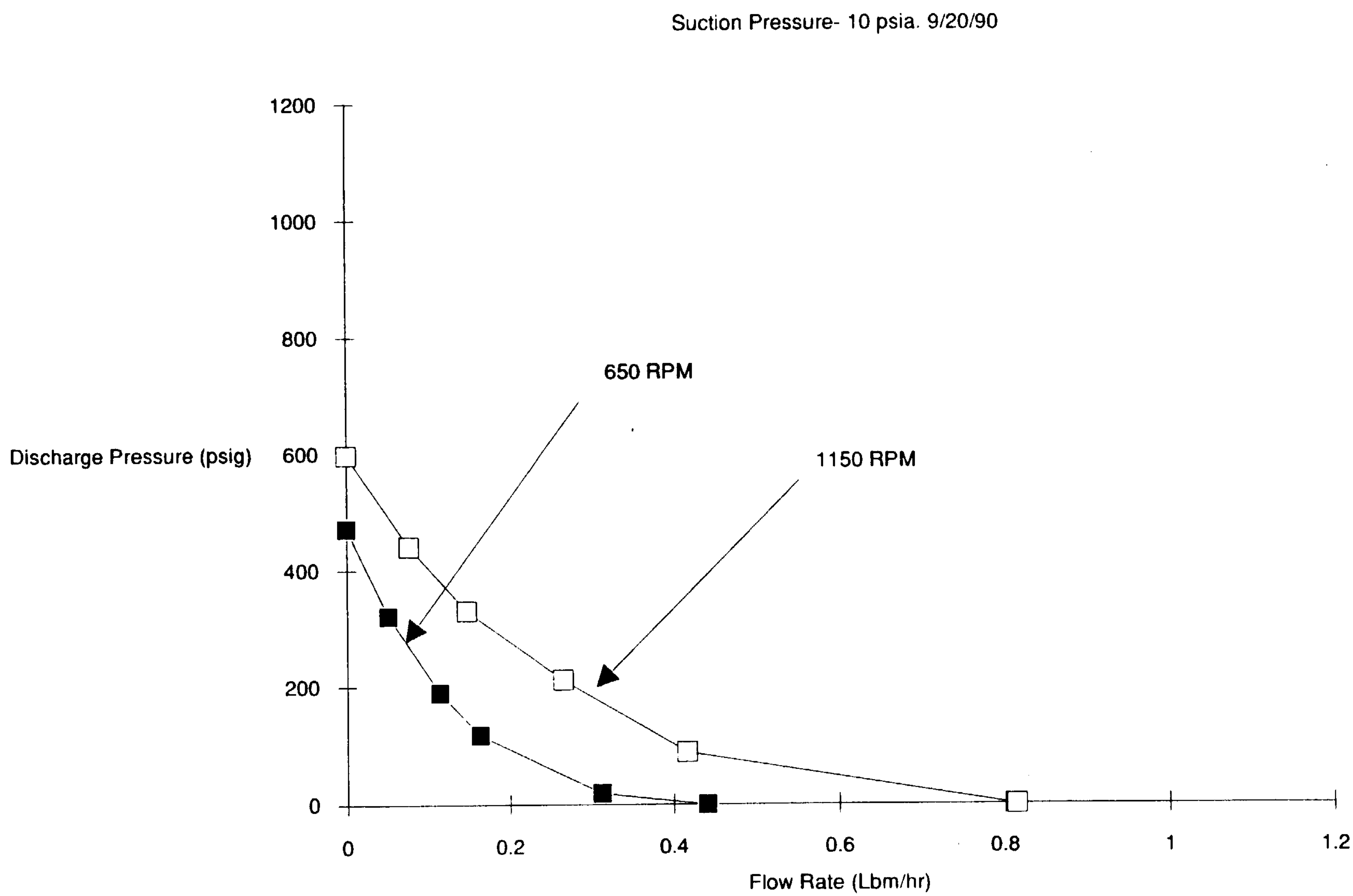


Figure 3. Flow Rate versus Discharge Pressure at 10 psia Suction Pressure

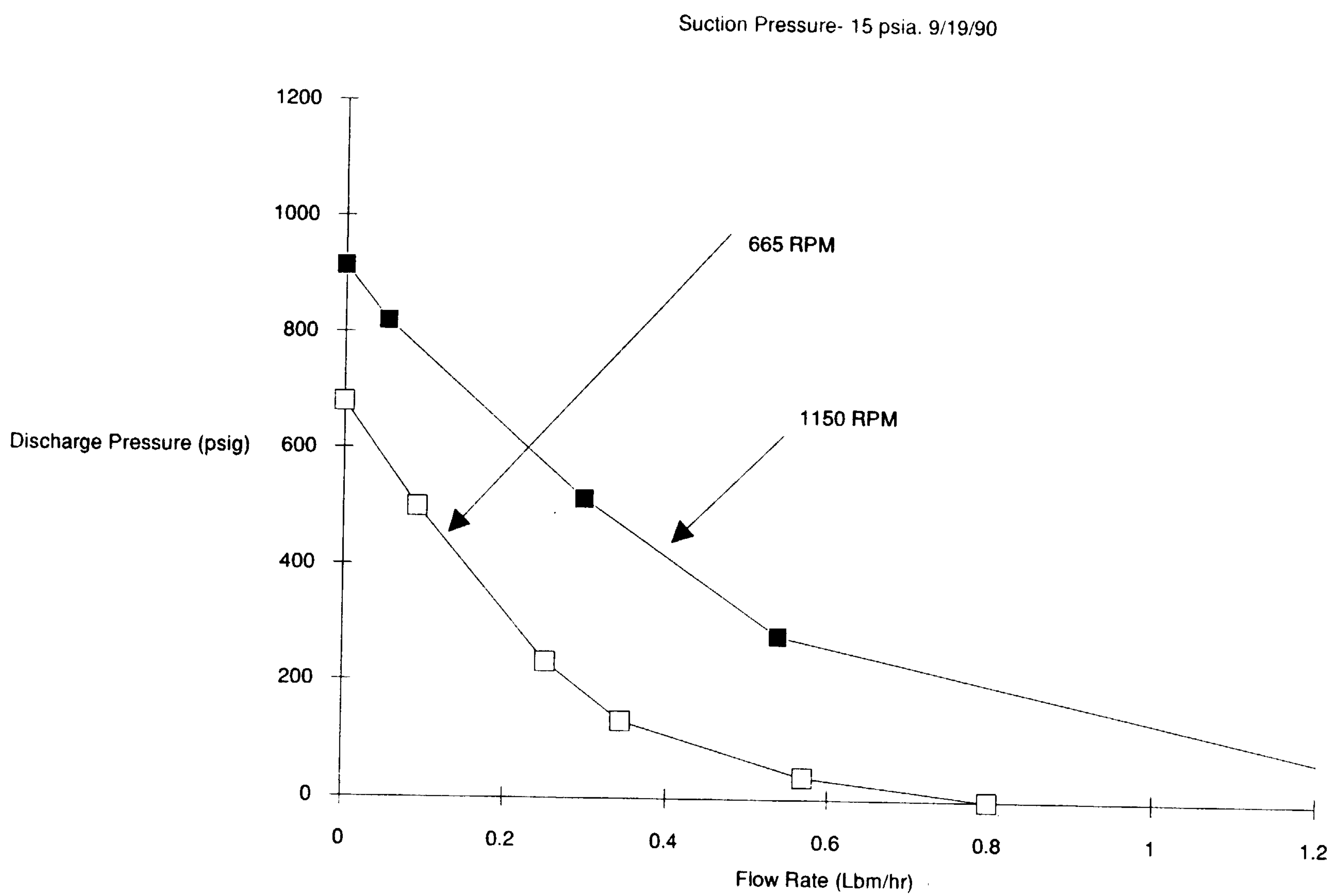


Figure 4. Flow Rate versus Discharge Pressure at 15 psia Suction Pressure

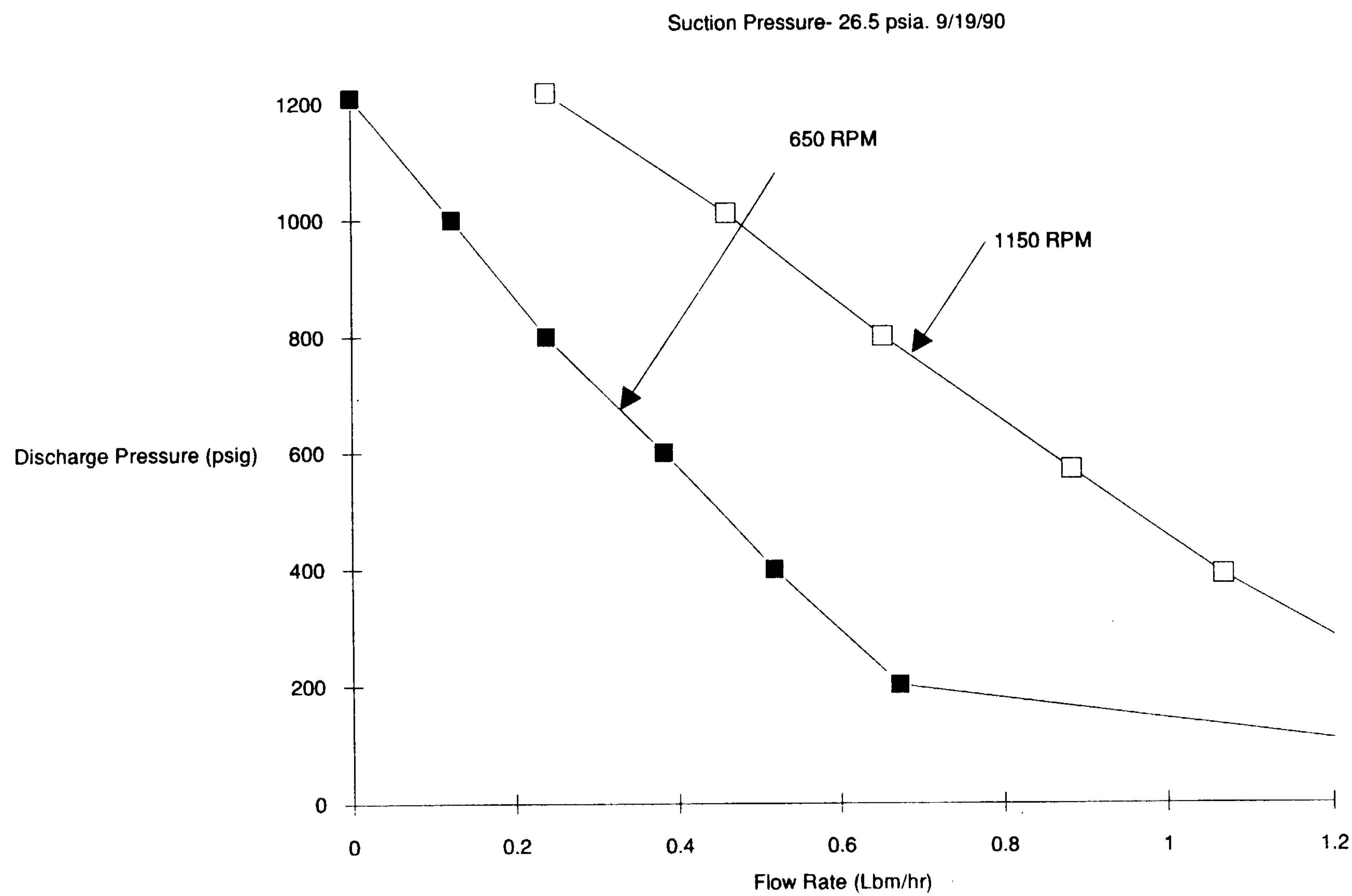


Figure 5. Flow Rate versus Discharge Pressure at 27 psia Suction Pressure

plots of flow rate versus discharge pressure for each different suction pressure. As expected, the plots show flow rate increases with decreasing discharge pressure. The discharge pressure and flow rate increase with increasing compressor speed. Because the compressor motor controller was being repaired when the above described tests had to be performed, an externally mounted motor was used to drive the compressor during these tests. After the motor controller was repaired, the performance curve shown in Figure 3 at 15 psia was rerun and there was no change in compressor performance.

The performance tests verify the compressor meets the required flow, pressure and temperature specifications. All of these tests were performed with the coolant fluid temperature at 20°C.

3.8 Operating Life Testing

3.8.1 Design Requirement: EIS 4.2.5.1 Operating Life

Minimum useful on-orbit operating life at the duty cycles specified in Section 4.2.5.2 shall be 10,000 operating hours. The compressors shall be refurbishable for a minimum of an additional 10,000 operational hours of on-orbit service.

3.8.2 Test Item

Subassembly Test Article.

3.8.3 Test Description

Because the compressor is designed to provide long life, it is impossible to base useful operating life predictions on short term (100 hour) tests (since little wear has occurred in this time). For this reason, the life tests outlined under Section 3.2 are used for life predictions discussed in Section 4.3.

3.8.4 Test Equipment

Equipment listed under Surface Wear Testing.

3.8.5 Test Results

The results of testing are given in Section 3.2 and life predictions are given in Section 4.3. In addition to the STA testing, long term testing of the compressor is planned at NASA-JSC. The results of this testing will provide a much more accurate life prediction.

4.0 VERIFICATION BY ANALYSIS

Table VI lists the EIS items that require verification by analysis. Verification by analysis is primarily used where simulated design conditions cannot be met, test data must be extrapolated beyond the test parameters, and where articles of similar design have been verified to equivalent requirements.

A description of the analysis objectives, analysis description, and analysis results for each EIS item to be verified by analysis is given below.

Table VI. Verification By Analysis

EIS Section No. & Title		Method of Verification
3.2.2.3	Surface Wear	Analysis & Test
4.2.2.3	Burst Pressure	Analysis
4.2.5.1	Operating Life	Test & Analysis

4.1 Surface Wear

4.1.1 Design Requirement: EIS 3.2.2.3 Surface Wear

The wear and attendant particle generation at any dynamically interfacing surfaces (contacting surfaces under relative motion) shall not introduce contaminant into the fluid flow path and shall not impair the function of that specific interface, the compressor as a whole, or the user system for the life of the compressor.

4.1.2 Analysis Description

Surface wear tests conducted on the Subassembly Test Article (STA) and the prototype compressor. The results of these limited duration tests were used to predict compressor operating life. The critical wear surfaces to be analyzed were the piston seal and guide rings, cam follower, compressor valves, and the bearings.

4.1.3 Analysis Results

Based on the results of wear tests on the STA and the prototype compressor, the wear and particle generation of the cam followers, bearings, and check valves are not significant. Wear products are generated by the piston seals and piston guide (the portion of the piston that contacts the cylinder wall). As part of the piston seal and guide "break-in," material is transferred to the cylinder wall and some particulate is generated. This particulate is primarily confined to the cylinder above the piston, the spring cavity, and some particulate migration into the case and through the

valves into the pulsation bottles. Since the wear products are inert, soft, self-lubricating materials (and the quantity generated is very small, that is, the wear rates are very low) the operation of the compressor is not impaired by their presence.

The one area of concern is the cylinder discharge check valves. If the wear products accumulate at this location in a non-uniform manner, they could possibly cause the valve to remain partially open and leakage could occur. The experience from the wear testing is that this has not been a problem. Some wear products are seen on and around the valve seats, but since the material is soft, it has not caused valve leakage. The high valve forces on the seat will cause these soft materials to flatten out and provide a good smooth seat face.

The wear products that move into the discharge pulsation bottles will not cause any blockage or interfere with the gas flow simply because the volume of the wear products is miniscule compared with the pulsation bottle volume.

4.2 Burst Pressure

4.2.1 Design Requirement: EIS 4.2.2.3 Burst Pressure

Burst pressure shall be 2.5 times the maximum operating pressure, approximately 17.3 MPa (2500 psia) as a minimum.

4.2.2 Analysis Description

The engineering design calculations were performed using conservative estimates to assure safe operations. Each part was analyzed assuming the required burst pressure of 2500 psi and the resultant load or stress was then compared to the strength of the material used. Three types of analysis were performed. The first analysis was for cylindrical pressure vessels using the standard Barlow Formula for pressure tubes where maximum sizes are equal to internal pressure times inner diameter divided by two times the wall thickness. The three parts analyzed in this manner are the pulsation bottles, compression cylinders and the crankcase housing. The pulsation bottles are simple cylinders internally with a more complex shape externally. The pressure vessel analysis used the minimum wall thickness of the part, but most of the cylinder walls are in excess of this minimum for mounting. The crankcase housing, however, is a complex part with a number of wall penetrations and threaded holes for end cap mounting. The selected wall thickness for all three vessels is between four and five times that required for a simple cylinder. The second analysis was for the end caps for the crankcase housing and pulsation bottles. These end caps/flanges were analyzed as flat, thin, circular plates. The third analysis was for the screws used to fasten the crankcase end caps, the cylinder flange to housing, and the cylinder flange to pulsation bottles.

4.2.3 Analysis Results

The Pressure Vessel Analysis Results are:

	Max. Calculated Stress	Yield Strength	Safety Factor *(for simple tube)*
Pulsation Bottles	8600 psi	40,000 psi	4.6
Compression Cylinders	7800 psi	40,000 psi	5.1
Crankcase Housing (equivalent cylinder)	8900 psi	40,000 psi	4.5

The Screw Bolt Strength Analysis Results are:

	Pressure Force	Screw Clamp Load	Screw Tensile Strength
Valve Plate/Pulsation Suct. (4 each, .250-28 UNF)	5185 lbs.	6000 lbs.	10,800 lbs.
Cylinder Flange/Housing (4 each, .250-28 UNF)	6447 lbs.	6000 lbs.	10,800 lbs.
Rear Flange End Cap (10 each, .375-24 UNF)	24,050 lbs.	36,200 lbs.	65,000 lbs.
Front Flange End Cap (8 each, .375-24 UNF)	22,365 lbs.	28,960 lbs.	52,000 lbs.

The Flange/Flat Plate Analysis Results are:

	Max. Calculation Stress	Yield Strength	Safety Factor*
Rear Flange End Cap	13,250 psi	40,000 psi	3
Front Flange End Cap	38,150 psi	40,000 psi	1.1

The pulsation bottle retainer ring force due to pressure was calculated to be 3712 lbs. with a material groove yield strength of 5458 lbs.

****Calculation neglects reinforcing webs. The actual part does have four webs which add an additional factor of safety.***

4.3 Operating Life

4.3.1 Design Requirement: EIS 4.2.5.1 Operating Life

Maximum useful on-orbit operating life at the duty cycles specified in EIS Section 4.2.5.2 shall be 10,000 operating hours. The compressors shall be refurbishable for a minimum of an additional 10,000 operational hours of on-orbit service.

4.3.2 Analysis Description

The minimal useful compressor life is limited by the surface wear of the piston seals. Seal wear tests are described in Section 3.2 and those results used to predict seal life.

4.3.3 Analysis Results

The seal life analysis is based on the methods given in the ASME Design Manual on PTFE Seals in Reciprocating Compressors [American Society of Mechanical Engineers, "Manual of Material Selection, Design and Operating Practices, PTFE Seals in Reciprocating Compressors," ASME, New York, NY 10017, 1975.] The calculation method is as follows:

Predicted Life ($T_{Pred.}$) in Hours

$$T_{Pred.} = \left(\frac{R_{limit}N}{P_m V_a}\right)_{projected} \times \frac{1}{K_{test}}$$

where:

$$K_{Test} = \left(\frac{RN}{P_m V_a T}\right)_{test}$$

R_{limit} = % loss of ring thickness

N = number of rings

$$P_m = \left(\frac{n}{n-1}\right) P_{suc}\left[(P_{dis}/P_{suc})^{\frac{n-1}{n}} - 1\right]$$

n = ratio of gas specific heats

V_a = stroke x RPM/6

The life prediction for the compressor is based on the STA seal wear test data presented in Figure 2 of Section 3.2 above. After the initial break-in period, the data shows a constant wear rate. If we use this wear rate and account for the material loss during the break-in period, the above procedure can be used to predict life. Based on assumed nominal pressure conditions, speeds, and 50% ring thickness loss, the predicted second stage seal life is 9500 hours. This predicted life, within the uncertainty of the STA data, meets the requirement for minimum compressor life.

5.0 VERIFICATION BY ASSESSMENT

Table VII lists the EIS items that will be verified by assessment. Verification by assessment requires the careful review and evaluation of design drawings or visual inspections. Verification of EIS requirements by the assessment method is commonly used for verification of surface finishes, tolerances, identification, and items requiring visual inspection.

A list of the EIS design requirement that will be verified by Assessment along with the assessment description is given below.

5.1 Mechanical

5.1.1 Design Requirement: EIS 3.1.1.2 Mechanical

The component shall be attached to a structure by bolting.

5.1.2 Assessment Results

Twelve tapped and threaded bolt holes on the compressor case provide mounting attachment.

5.2 Electrical

5.2.1 Design Requirement: EIS 3.1.1.3 Electrical

The component shall interface electrically through connectors meeting specification requirements found in EIS Section 2.0.

5.2.2 Assessment Results

Since the compressor motor and controller are not being flight qualified, and they are the only electrical components, this requirement is not applicable.

5.3 Lubrication

5.3.1 Design Requirement: EIS 3.1.2.2 Lubrication

Minimization of lubricants is a design objective. The lubricants used shall comply with EIS Section 3.3.1.5.

Table VII. Verification By Assessment

EIS Section No. & Title		Method of Verification
3.1.1.2	Mechanical	Assessment
3.1.1.3	Electrical	Assessment
3.1.2.2	Lubrication	Assessment
3.2.3	Safety, Reliability and Quality Assurance	Assessment

3.2.4.1	Transportation	Assessment
3.2.4.2	Storage in Protected Areas	Assessment
3.2.5	Transportability	Assessment
3.3.1.1	Materials and Processes	Assessment
3.3.1.2	Prohibited Materials	Assessment
3.3.1.3	Fluids	Assessment
3.3.1.4	Material Compatibility	Assessment
3.3.1.5	Lubricants	Assessment
3.3.1.6	Dissimilar Metals	Assessment
3.3.1.7	Platings and Castings	Assessment
3.3.1.8	Protective Treatment	Assessment
3.3.1.11	Non-Destructive Evaluation	Assessment
3.3.1.12	Cleanliness	Assessment
3.3.1.15	Assembly Cleanliness	Assessment
3.3.1.16	Parts Standardization	Assessment
3.3.1.17	Threads and Fasteners	Assessment
3.3.1.18	Locking Threaded Parts	Assessment
3.3.1.19	Prohibited Retaining Methods	Assessment
3.3.1.20	Surface Texture	Assessment & Test
3.3.1.21	Dimensioning and Tolerancing	Assessment
4.2.1	Fluid	Assessment
4.2.7	Line Sizes	Assessment

5.3.2 Assessment Results

The only lubricant used in the compressor was in the crank main bearings and the actuator bearings. No other lubricants were used.

5.4 Safety, Reliability, and Quality Assurance

5.4.1 Design Requirement: EIS 3.2.3 Safety, Reliability, and Quality Assurance

The safety, reliability, and quality assurance provisions for the components shall be tailored from NHB-5300.04 (1D-2).

5.4.2 Assessment Results

The safety, reliability, and quality assurance provisions are addressed in the Failure Modes and Effects Analysis (FMEA) presented in the Final Design Report.

5.5 Transportation

5.5.1 Design Requirement: EIS 3.2.4.1 Transportation

The compressor must be able to survive the environmental extremes encountered during transportation. The EIS Section 3.2.4.1 outlines the environmental conditions encountered for both air and ground transportation. Exposure to these conditions shall not result in damage, deterioration, or otherwise impair the capability of the component to meet the compressor operating performance requirements.

5.5.2 Assessment Results

Each compressor component can withstand the environmental conditions outlined in EIS 3.2.4.1 since the compressor operating conditions (of pressure, temperature, vibration, ...) are much more severe than the transportation environment.

5.6 Storage in Protected Areas

5.6.1 Design Requirement: EIS 3.2.4.2 Storage in Protected Areas

The compressor must survive exposure to the environment encountered during storage without damage or deterioration. The environmental conditions are listed under EIS 3.2.4.2.

5.6.2 Assessment Results

Each compressor component can withstand the environmental conditions outlined in EIS 3.2.4.2 since the compressor operating conditions are much more severe than the storage environment.

5.7 Transportability

5.7.1 Design Requirement: EIS 3.2.5 Transportability

The compressor shall be designed to be capable of being handled and transported to user facilities without damage or degradation while utilizing available methods of transportation with the item prepared for shipment in accordance with EIS Section 7.0 requirements. The equipment design shall be compatible with the planned packaging and transportation system to the extent that loads induced in the equipment during transportation shall not produce stresses, internal loads, or deflections resulting in damage to the equipment.

5.7.2 Assessment Results

The compressor design requirement dictated a strong/durable compressor design that would preclude damage from loads encountered in handling and transportation.

5.8 Materials and Processes

5.8.1 Design Requirement: EIS 3.3.1.1 Materials and Processes

Materials and processes for the compressor shall be selected in accordance with SE-M-0096, JSC-08962-U, JSC-09604-B, JSC-30233, and NHB-1014.

5.8.2 Assessment Results

Material selection during the prototype compressor design were based on a review of loading, environmental considerations, and life expectancy. JSC-09604-B was utilized to screen candidate materials for safety hazard/contamination problems. Since the prototype compressor is not intended for flight qualification, all of the above specifications are not applicable.

5.9 Prohibited Materials

5.9.1 Design Requirement: EIS 3.3.1.2 Prohibited Materials

The following materials are prohibited from use unless specifically approved by the Government:

(a) Cadmium, Zinc, or selenium except internal to hermetically sealed devices.

(b) Unalloyed, electro-depositioned tin unless subsequently fused or reflowed.

(c) Corrosive solder fluxes unless detailed cleaning procedures are specified along with appropriate verification methods to ensure removal of residual contaminants.

(d) Mercury and compounds of mercury.

(e) Teflon, vinyl, and polyvinylchloride as insulation for electrical hookup wiring. (Does not apply to Teflon insulated coaxial cables.)

(f) Materials which exhibit natural radioactivity such as uranium, radium, thorium and/or any alloys thereof.

5.9.2 Assessment Results

The above listed materials are not utilized in the compressor.

5.10 Fluids

5.10.1 Design Requirement: EIS 3.3.1.3 Fluids

Procurement and use of fluids shall be controlled to the extent specified in SE-S-0073 unless otherwise specified.

5.10.2 Assessment Results

Fluids used in the prototype compressor manufacturing and testing do not require any special control.

5.11 Material Compatibility

5.11.1 Design Requirement: EIS 3.3.1.4 Material Compatibility

Materials and any lubricants used in the construction of the compressor shall be suitable for use with the fluids as specified in Section EIS 4.0 at the temperatures and pressures defined in that section.

5.11.2 Assessment Results

The materials and lubricants selected for use in the compressor are compatible with the conditions stated in EIS 4.0. A material compatibility study for the compressor components is presented in the Prototype Final Design Report.

5.12 Lubricants

5.12.1 Design Requirement: EIS 3.3.1.5 Lubricants

Minimization of lubricants is a design objective. The compressor designs may employ the use of lubricants provided that they comply with the following for the life of the component:

(a) Meet the requirements in EIS 3.3.1.

(b) Do not introduce contamination by entering the fluid flow path.

(c) Are not lost and/or degraded as a result of exposure to the working fluids, exposure to the environments of EIS 3.2.4, or from the operation of the component over its entire life, such that the ability of the component to meet the requirements specified herein is impaired.

5.12.2 Assessment Results

This specification will be assessed under EIS 3.1.2.2 (Section 5.4).

5.13 Dissimilar Metals

5.13.1 Design Requirement: EIS 3.3.1.6 Dissimilar Metals

When dissimilar metals are used for parts that come in contact with each other, the materials selected shall comply with MIL-STD-889.

5.13.2 Assessment Results

The only dissimilar metals used in the prototype are stainless steel and high carbon steel that are in contact with anodized aluminum. Since the anodized aluminum is an insulator, these dissimilar metals pose no problems.

5.14 Platings and Castings

5.14.1 Design Requirement: EIS 3.3.1.7 Platings and Castings

The plating of materials which will be in contact with operational fluids are restricted from use. The use of castings is prohibited.

5.14.2 Assessment Results

No casting will be employed in the compressor. For the prototype compressor, some platings (on bearing races) are employed for availability reasons. These will be eliminated in the final design of the flight hardware.

5.15 Protective Treatment

5.15.1 Design Requirement: EIS 3.3.1.8 Protective Treatment

The use of any protective coating that will chip, crack, abrade, peel, or scale with usage, age, or extremes of climatic and environmental conditions is restricted from use.

5.15.2 Assessment Results

No protective coatings that will chip, crack, abrade, peel, or scale are used in the compressor.

5.16 Non-Destructive Evaluation

5.16.1 Design Requirement: EIS 3.3.1.11 Non-Destructive Evaluation

The contractor shall consider the development and potential use of non-destructive evaluation inspection techniques in the design and construction of the compressor.

5.16.2 Assessment Results

No destructive evaluation methods were required.

5.17 Cleanliness

5.17.1 Design Requirement: EIS 3.3.1.12 Cleanliness

Significant surfaces of the compressor shall be cleaned to Level 100A.

5.17.2 Assessment Results

This requirement is verified under EIS 3.3.1.13.

5.18 Assembly Cleanliness

5.18.1 Design Requirement: EIS 3.3.1.15 Assembly Cleanliness

Compressor piece parts shall be individually cleaned prior to assembly and maintained clean during the assembly process. The entire assembly shall be verified cleaned at the completion of assembly.

5.18.2 Assessment Results

This requirement is verified under EIS 3.3.1.13.

5.19 Parts Standardization

5.19.1 Design Requirement: EIS 3.3.1.16 Parts Standardization

Standardization parts utilization shall be based upon:

(a) Selection of qualified parts.

(b) Proper derating and application.

(c) Minimizing the number of parts.

5.19.2 Assessment Results

Very few standardized parts are employed in the prototype compressor. The only standardized parts are fasteners, snap rings, and O-rings.

5.20 Threads and Fasteners

5.20.1 Design Requirement: EIS 3.3.1.17 Threads and Fasteners

Screw threads shall be in accordance with FED-STD-H28.

5.20.2 Assessment Results

All of the threaded fasteners used in the prototype compressor are 304SS material and standard thread patterns.

5.21 Locking Threaded Parts

5.21.1 Design Requirement: EIS 3.3.1.18 Locking Threaded Parts

Threaded parts shall be positively locked. Preferred locking methods, in order of preference, are as follows:

(a) Safety wiring in accordance with MS-33540.

(b) Self-locking nuts.

(c) Castellated nuts and cotter pins.

(d) Screw locking screw thread inserts.

(e) Self-locking bolts or screws and lock washers.

5.21.2 Assessment Results

Self-locking Helicoil inserts and Spiralock self-locking tapped threads were used throughout the compressor with the exception of the valve retainers which do not have a positive locking method. This was done in the prototype to allow disassembly and reassembly for inspection and parts replacement. Many threaded fasteners would be eliminated (by welding the parts together) in flight hardware.

5.22 Prohibited Retaining Methods

5.22.1 Design Requirement: EIS 3.3.1.19 Prohibited Retaining Methods

Staking, press fits, or crimping shall not be used as a primary means of retaining detail parts or subassemblies.

5.22.2 Assessment Results

No components on the compressor require staking, press fits, or crimping.

5.23 Surface Texture

5.23.1 Design Requirement: EIS 3.3.1.20 Surface Texture

Surface texture limitations shall be in accordance with ANSI B46.1-78.

5.23.2 Assessment Results

Surface texture limitations are in accordance with ANSI B46.1-78.

5.24 Dimensioning and Tolerancing

5.24.1 Design Requirement: EIS 3.3.1.21 Dimensioning and Tolerancing

All dimensioning and tolerances shall be in accordance with ANSI Y14.5M-82 and DOD-STD-100.

5.24.2 Assessment Results

Dimensioning and tolerancing are in accordance with ANSI Y14.5M-82 and DOD-STD-100.

5.25 Fluid

5.25.1 Design Requirement: EIS 4.2.1 Fluid

The bulk working fluid will consist of an oxidizing gas mixture as specified in Appendix 2, paragraph 1.2 in the EIS. Trace contaminants potentially mixed with the bulk gas mixture are specified in Appendix 2, Section 2.0 in the EIS. The compression of this gas mixture presents the following potential phase change or chemical reaction issues:

(1) condensation of CO_2 at low temperatures (below 0°F and 300 psia)

(2) a reaction between the fuels (C_2H_2, NH_3, C_2H_4, C_3H_8, etc.) and oxygen at elevated temperatures.

5.25.2 Assessment Results

Because the quantity of CO_2 in the gas is so low (1.9%), and the compressor clearance volume is relatively large, the condensation of CO_2 will not be a problem. The presence of a small amount of liquid in the compressor cylinder will not present any major problem. The compressor will also not operate below 0°F.

The chemical reaction outlined above would be a concern if the "fuel" gas concentration was significant. Because the "fuel" gas concentration is less than 2%, the effect of a reaction during compression would have no major effect on the compressor operation.

5.26 Line Sizes

5.26.1 Design Requirement: EIS 4.2.7 Line Sizes

Line Sizes are as follows:

(a) Inlet: 0.01 m (3/8 inches)

(b) Outlet: 0.01 m (3/8 inches)

5.26.2 Assessment Results

The compressor line sizes are 3/8 inch.

6.0 CONCLUSIONS

The verification program has documented that the on-orbit compressor prototype meets the requirements of the End Item Specification (EIS) relevant to prototype hardware. The prototype compressor is 3/8 of the allowable weight (30 lbs. versus 80 lbs.), 1/3 of the allowable volume (0.5 cu. ft. versus 1.5 cu. ft.), and 1/2 of the allowable power (500 watts versus 1000 watts). The performance requirements of flow rate, discharge pressure, and suction pressure were independently verified. At a suction pressure of 27 psia and a compressor speed of 650 RPM, the prototype

developed a deadhead pressure of 1210 psia, a 0.124 LBm./hr. flow rate at a 1000 psia discharge, a nominal flow rate of 0.24 LBm./hr at 800 psia discharge and 1.827 LBm/hr. flow rate at zero pressure rise. This performance clearly meets the EIS requirements of an outlet pressure range of 100 to 1000 psia (1200 psia maximum), and a nominal flow rate of 0.25 LBm/hr (1.1 LBm/hr. maximum). The one area of marginal performance is the life of the second stage piston seals. With the current Space Station interest in significantly longer life than the EIS requirement of 10,000 hours, we recommend that this area be further developed. We specifically recommend that a lubricated seal ring technology be developed for significant life extension.

The overall conclusion is that we have developed compressor technology for on-orbit applications. This technology balances all of the complex design requirements and is provided within a time frame consistent with the support of the Space Station Fluid Systems Development. The verification program has documented that the performance of the prototype waste gas compressor does indeed meet the EIS requirements.